AUTOMOBILE ENGINEER

DESIGN · PRODUCTION · MATERIALS

Vol. 49 No. 7 JULY 1959 PRICE: 3s. 6d.

MIDCYL RESEARCH also helps smooth out problems

like the potter gently smoothing out his clay
Midcyl research helps smooth the way of the
Auto Engineer with such of his problems as are
associated with Cylinder Blocks, Cylinder Heads,
Camshafts and Brake Drums

MIDCYL

THE MIDLAND MOTOR CYLINDER CO. LTD., SMETHWICK, STAFFS

H-W.80

"BRAKES FULL ON"

The job of this constant-torque machine in the B.B.A. Research and Development Laboratories is to test the wearing properties of Mintex Brake Liners. Its effect is similar to that of driving a car with the brakes full on. The pressure on the four test samples is varied inversely to the coefficient of friction to give a constant power absorption. The test is continued until 0.1" is worn away. This is one of the many exacting tests to which Mintex materials are subjected in the continuous programme for the development of higher braking standards.

you can rely on

MINTEX

Mintex Brake and Clutch Liners are manufactured by British Belting & Asbestos Limited, Cleckheaton, Yorkshire, and are available from MINTEX Service Depots and Distributors throughout the country.

Giant trucks link Sahara to the Sea

Atlas Copco air tools speed production of the Berliet heavyweight

The T100 is a giant truck built by Berliet of France for a giant task. It will carry vital drilling and building equipment into the depths of the Sahara for the newly found oil fields. Fitted with an air conditioned cabin, 300 km. radio range and low pressure tyres, this new addition to the Berliet desert fleet moves over the sand as surely as a camel.

Atlas Copco's contribution

Throughout the Berliet factories Atlas Copco air tools are in use. Drills, tappers and grinders are hard at work on cylinder heads, crank shafts, lubrication systems and bodywork. Their efficient, economical performance is helping Berliet to maintain their swift rate of production—every ten minutes another vehicle comes off the line!

Like Berliet in France, industries in all parts of the world are finding that Atlas Copco pneumatic tools are an aid to quick production at lower cost.

A complete range of compressed air equipment

Atlas Copco manufactures portable and stationary compressors, rock-drilling equipment, loaders, pneumatic tools and paint-spraying equipment. Sold and serviced by companies or agents in ninety countries throughout the world.

Atlas Copco PUTS COMPRESSED AIR TO WORK FOR THE WORLD

Contact your local Atlas Copco company or agent or write to Atlas Copco AB, Stockholm 1, Sweden, or Atlas Copco (Great Britain) Limited, Maylands Avenue, Hemel Hempstead, Herts.

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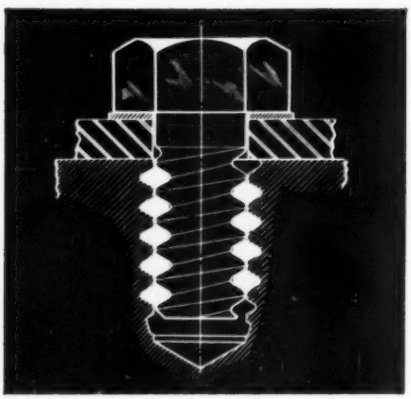

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screw thread inserts

Heli-Coil Inserts are self-anchoring thread liners made from high tensile stainless steel wire. In tapped holes they provide a conventional thread with higher loading strengths and greater resistance to wear and stresses than unprotected threads. The Heli-Coil eliminates stripping, seizing, galling and corrosion. It literally *armours* the thread. The Heli-Coil offers a unique opportunity for product cost revision. It is "a natural" for automation, it can make dramatic cuts in time and labour costs. Assembly is the ultimate in simplicity — just drill, tap and install. The Heli-Coil saves weight and space. It improves the serviceability and appearance of the end product. We suggest you write soon for data on Heli-Coil, the British-made Insert that is available internationally. It is a product of the day and this atomic age.

For further details write for sales leaflet APL. 48/E2.

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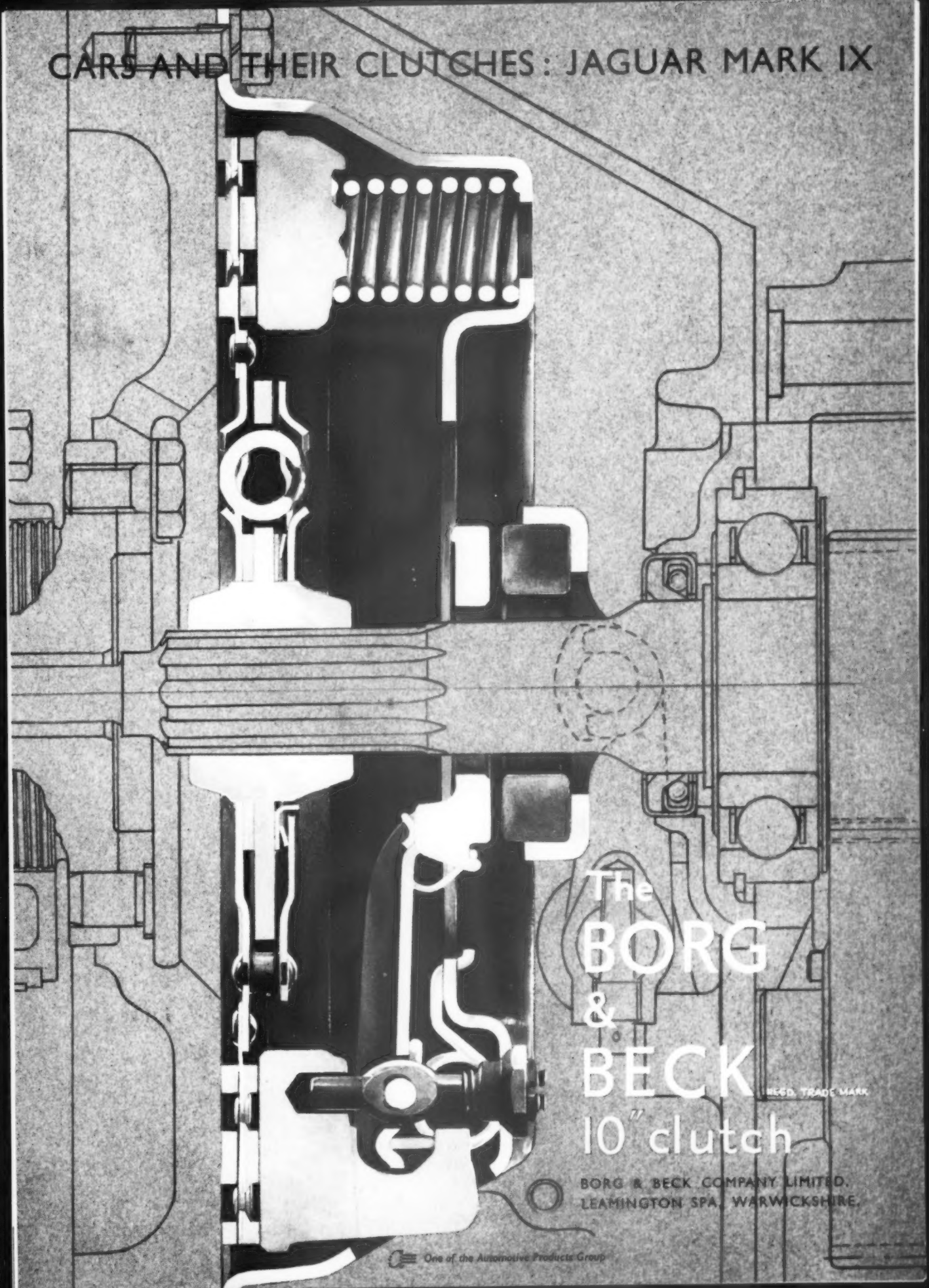
* *HELI-COIL is a registered trade mark*

CARS AND THEIR CLUTCHES: JAGUAR MARK IX



The BORG & BECK
REGD. TRADE MARK
10" clutch

BORG & BECK COMPANY LIMITED,
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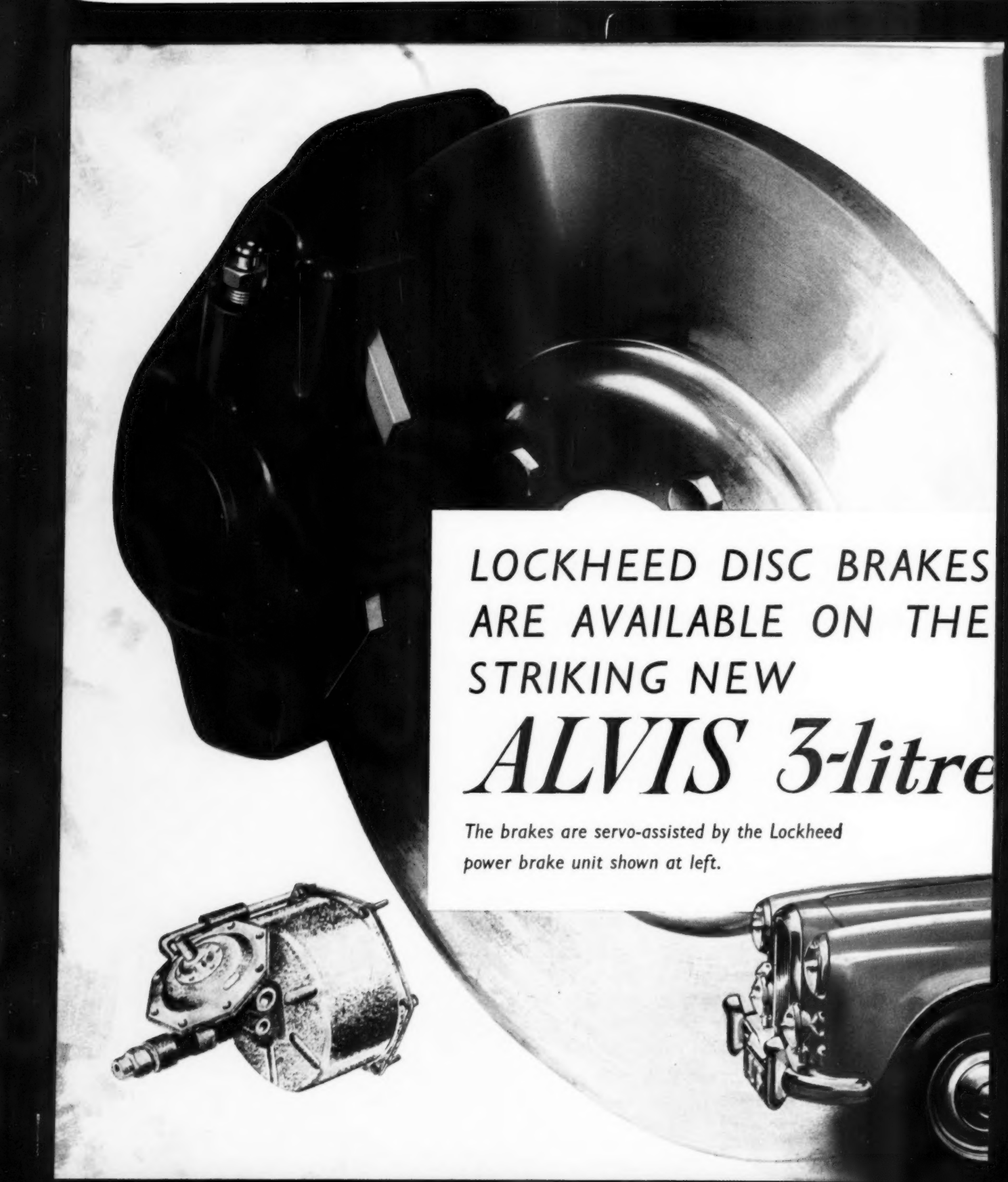
One of the Automotive Products Group



LOCKHEED DISC BRAKES ARE AVAILABLE ON THE STRIKING NEW

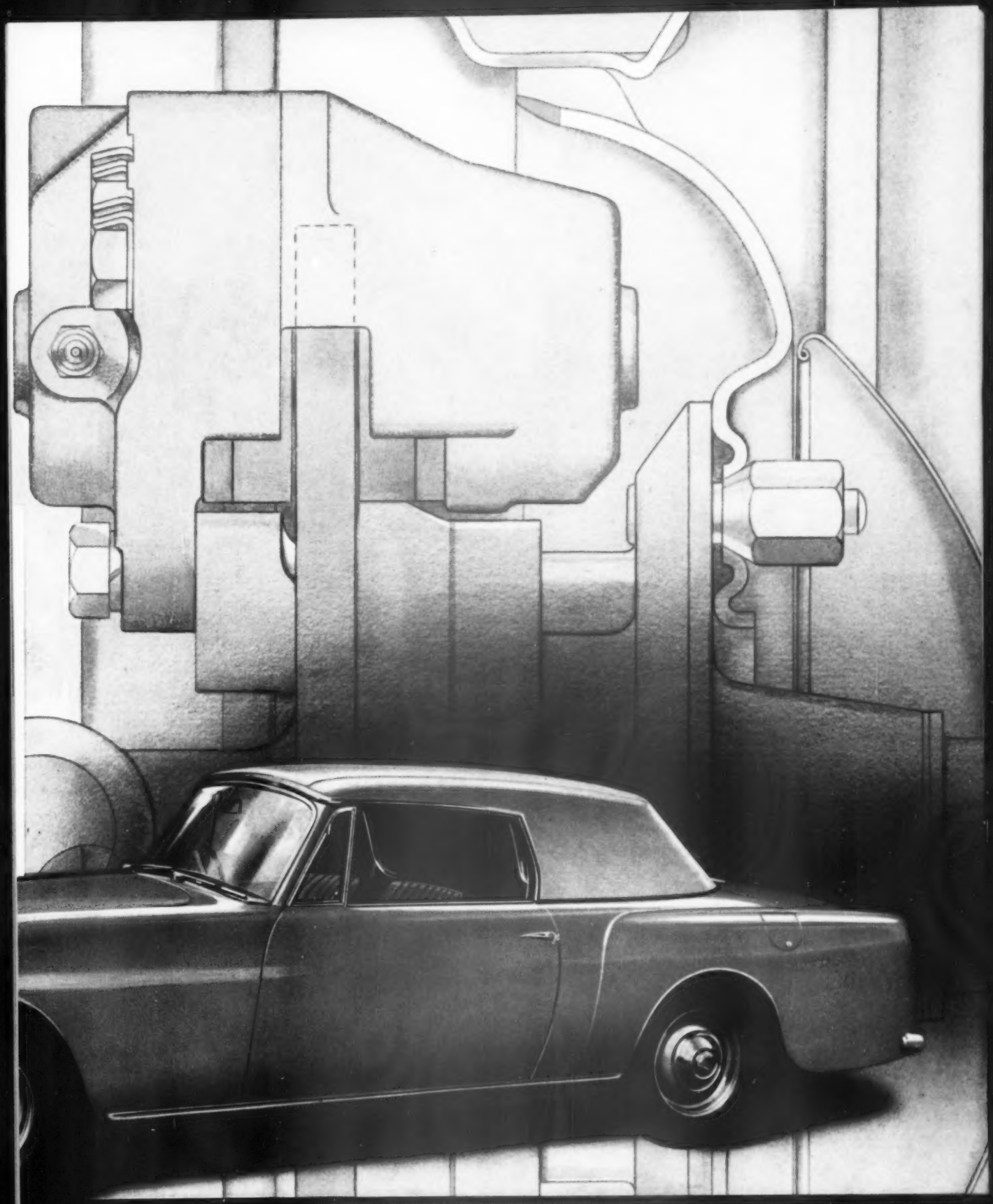
ALVIS 3-litre

The brakes are servo-assisted by the Lockheed power brake unit shown at left.

LOCKHEED

REGD. TRADE MARK LOCKHEED

LOCKHEED HYDRAULIC BRAKE COMPANY



DISC BRAKES

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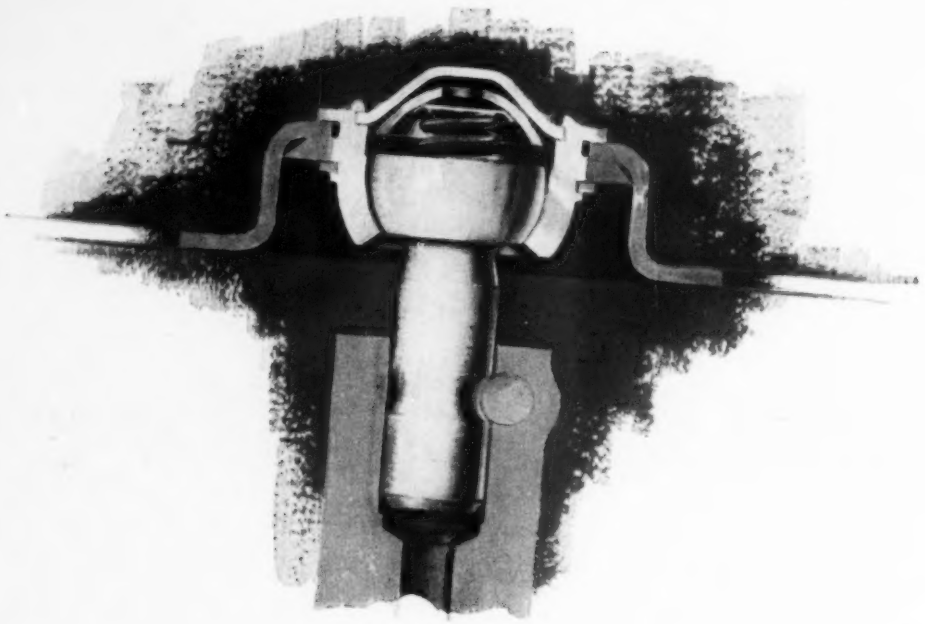
UPPER SUSPENSION JOINT



ON THE VAUXHALL 'VICTOR'

The sectional sketch shows the simple Thompson ball joint as supplied by us; this performs a double function, serving as steering swivel and at the same time providing the articulation between the swivel post and the top wish bone.

As in all Thompson steering and suspension joints, this is self-adjusting.

The seal and other details are provided by Vauxhall Motors Ltd.

Thompson

SELF-ADJUSTING

SUSPENSION AND STEERING JOINTS

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NEW
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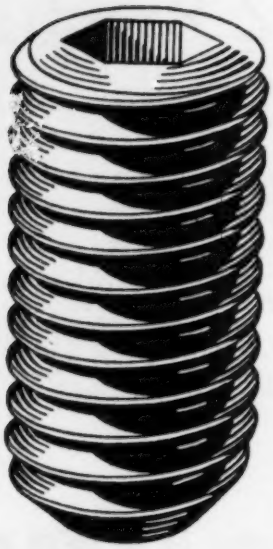

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which governs the making of

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and uniformity
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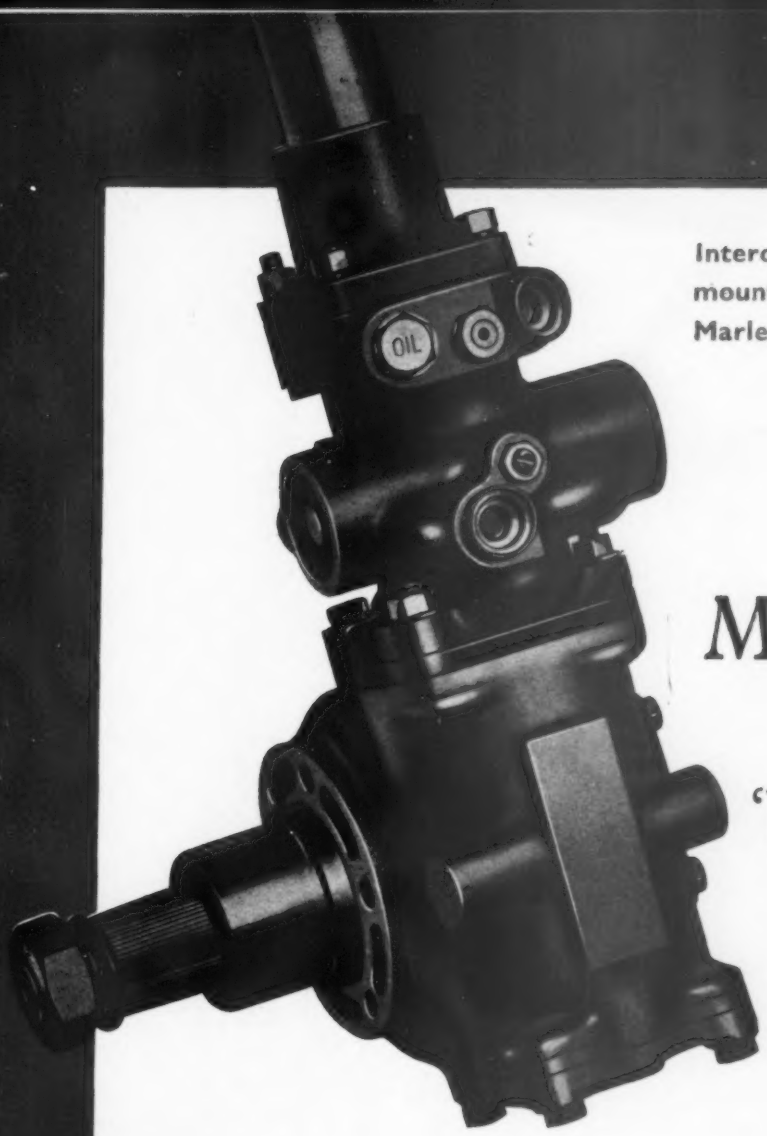

Interchangeable, on the same mounting, with the standard Marles manual gear unit.

MARLES type 4 'Universal' power-steering gear

This latest type 4 unit embodies the wide experience of Marles in power-steering, and consists of a 2-valve box mounted on the top of the standard Marles 861 manual gear unit.

The type 4 Universal unit shown above, the type 3 Universal unit and the standard manual unit are all interchangeable on the same mountings.

Further particulars will be sent on request.

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THE AERODROME, WOODLEY, Near READING.

Sole proprietors of the Marles Steering Company Ltd.

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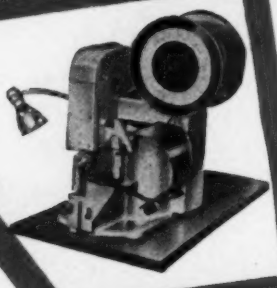
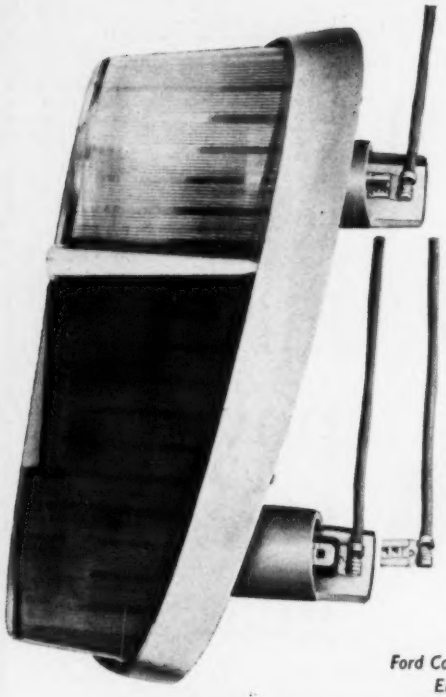
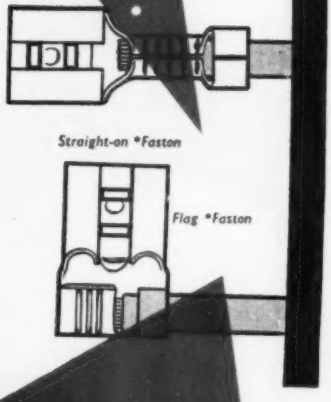


the means . . .



Thames Van rear lamp assembly



*Straight-on *Faston*

*Flag *Faston*

Ford Consul de Luxe — Electrical system uses both straight-on and flag Fastons*

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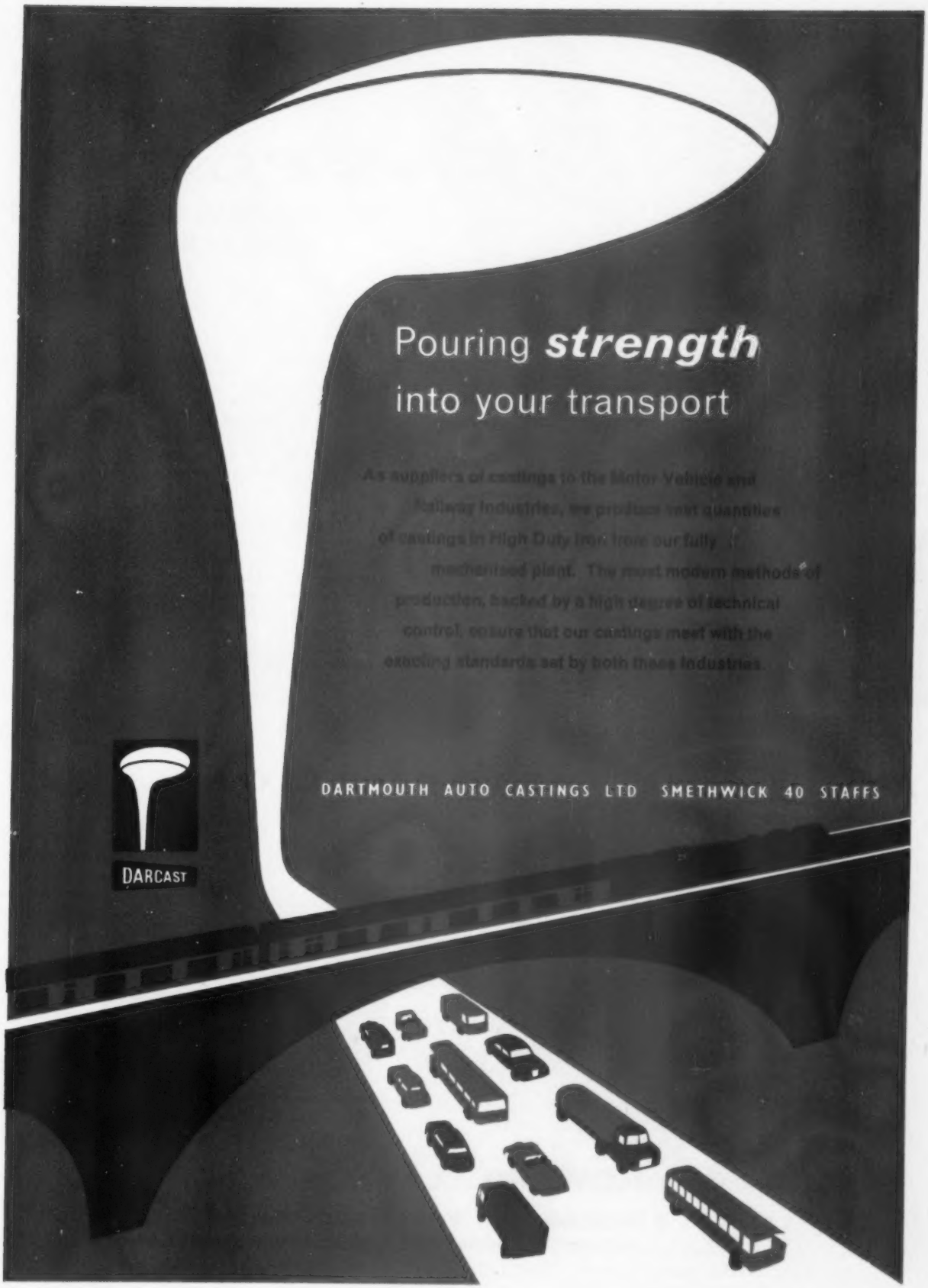
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into your transport
As suppliers of castings to the Motor Vehicle and Railway Industries, we produce vast quantities of castings in High Duty Iron from our fully mechanised plant. The most modern methods of production, backed by a high degree of technical control, ensure that our castings meet with the exacting standards set by both these industries.
DARCAST
DARTMOUTH AUTO CASTINGS LTD SMETHWICK 40 STAFFS

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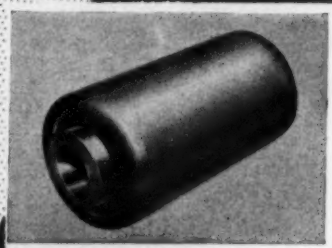
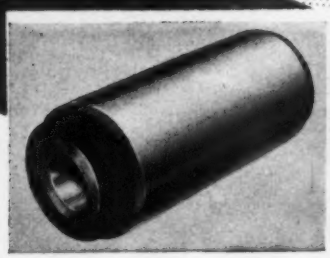
FLEXIBLE BEARINGS

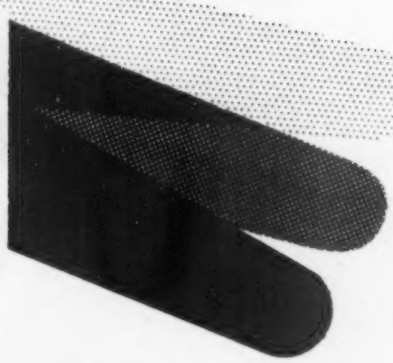


The best of both worlds is not easy of attainment, although it is constantly sought. We believe that the Clayflex Principle represents the best of both flexible bearing worlds in that it combines the advantages of controlled rubber deformation (resulting in the well-tried pressure-bond), with the added security of chemical bonding at the interface of rubber and inner member.

The wide range of types available ensures that every flexible bearing application can be catered for. Where axial control is particularly important the peened outer member or the extended rubber type are available, while for applications requiring the bearing to function as a torsional spring the B.S. type, which is fully chemically bonded without sacrifice of essential pre-load upon the rubber is recommended.

A CLATONRITE PRODUCTION





A fully illustrated brochure is available, and requests for copies are welcomed from Chief Technicians and Buyers.

Manufactured by

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Extra strong in design and construction this tool incorporates longer normal working life with minimum maintenance.

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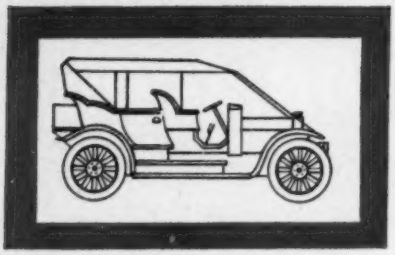

Sterling achievements span the history of an era

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The continuing success of Sterling Metals is due to their ability to supply castings of the highest quality in the desired quantities at the right time.

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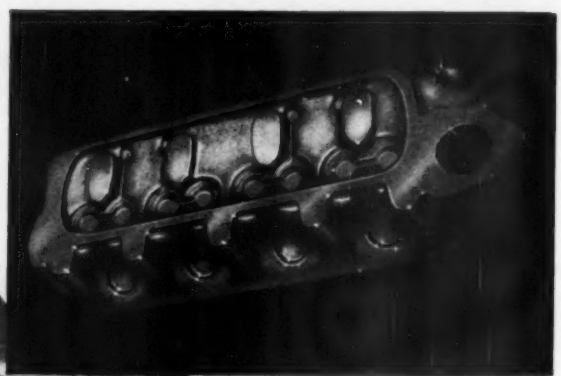

Cast iron cylinder head
for Austin A35 car.
By courtesy of Austin Motor Co Ltd.

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Expensive machine tools deserve a machine vice of equal precision—a role befitting the precision built DOWN-GRIP vice. Laborious setting-up time is substantially reduced. The quick action jaw slides right up to the job—a slight turn of the screw then gives instant and positive downward grip!

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Vice swivels on base through 360°

Reversible jaw plates for irregular shapes.

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✳ THE DORMER

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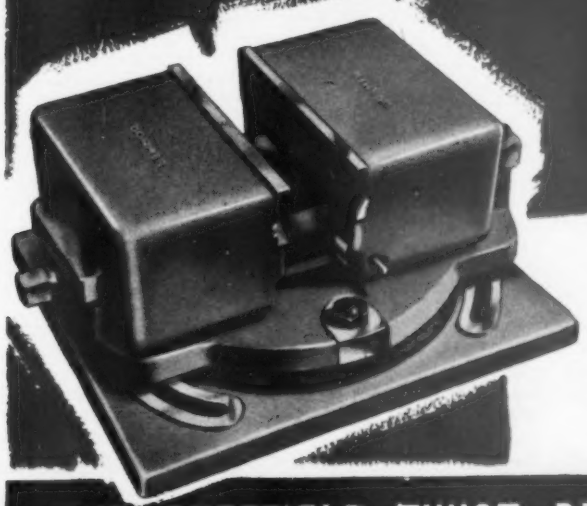

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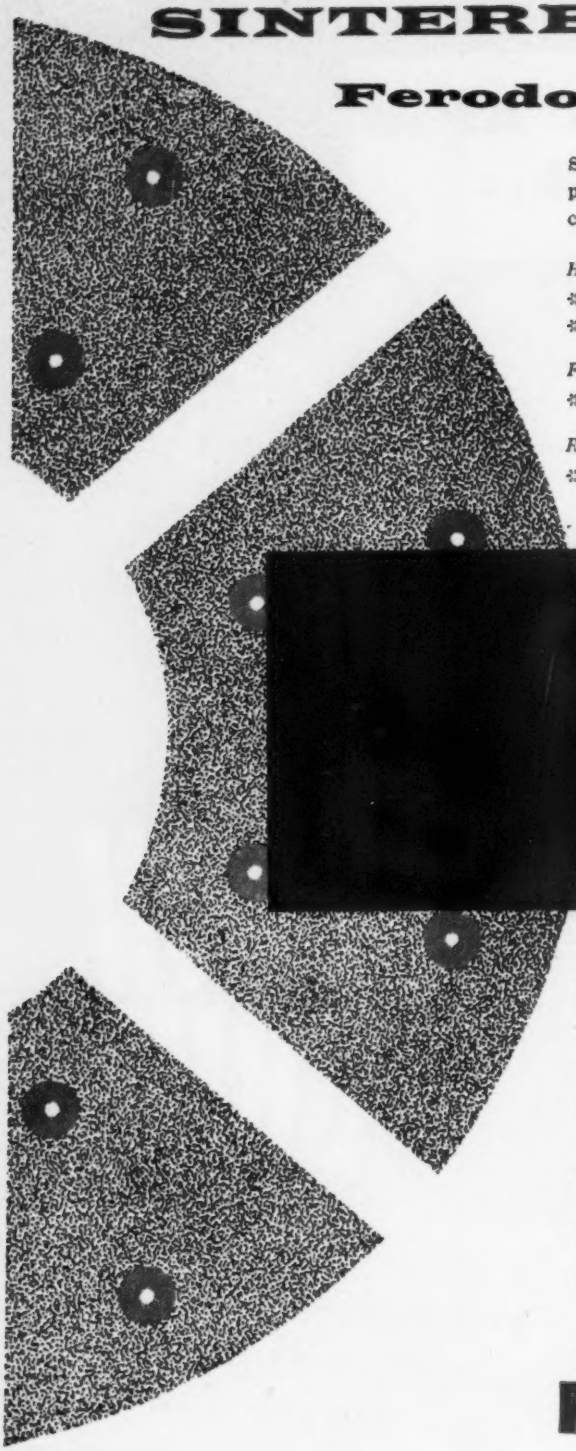

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Ferodo Sintered Metals withstand

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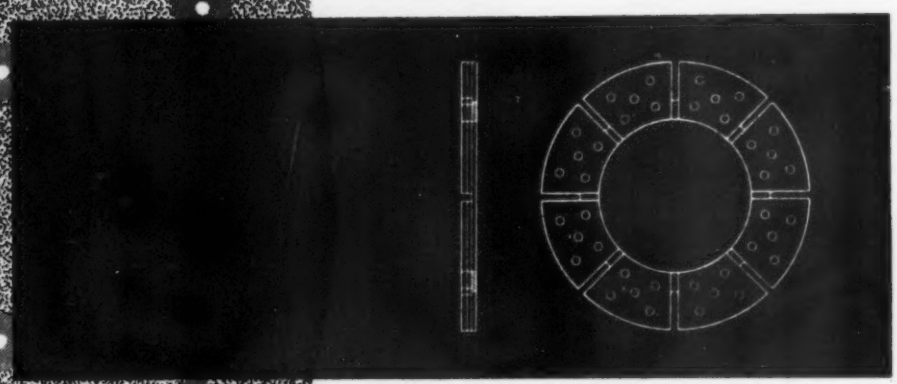
Reduced ratio of static to dynamic friction coefficients leads to

- ✲ SMOOTHER ENGAGEMENT



Ferodo Sintered Metal Types

SM.1 General purpose material for dry and oil immersed applications

SM.2 For oil immersed applications requiring smooth, gradual engagement such as automotive automatic transmissions

SM.3 Specially suited to dry, heavy duty applications such as engine master clutches on trucks and on tractor steering clutches

SM.4 Is intended as a mating surface for use against other Ferodo sintered metals to protect costly parts from heat damage and wear

SM.5 For oil immersed applications involving long slip periods such as seals and tension controlled clutches

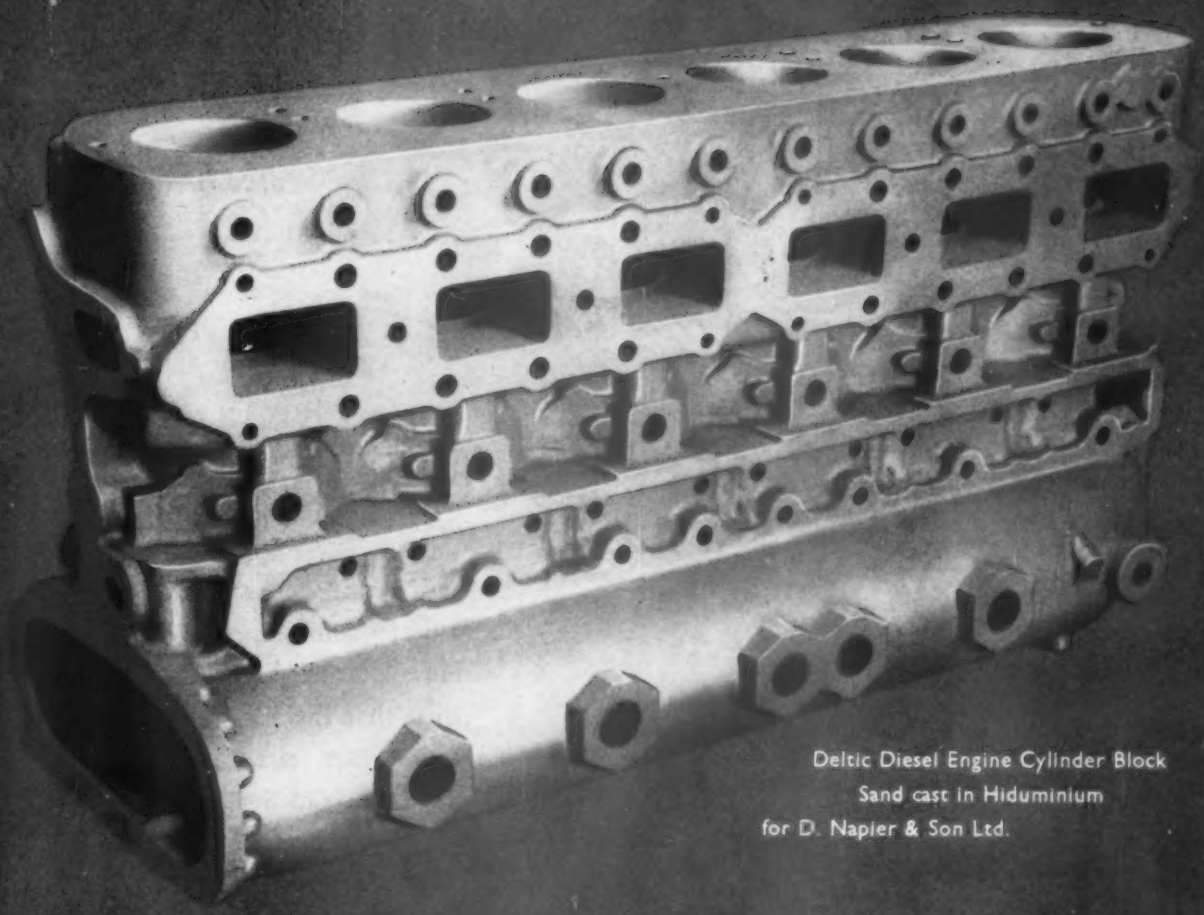
SM.6 For oil immersed applications where smooth engagement must be coupled with relatively high rating

Write to us for further technical information and brochure

FERODO

Sintered Metal Clutch Facings

FERODO LIMITED · CHAPEL-EN-LE-FRITH *A Member of the Turner & Newall Organisation*



Deltic Diesel Engine Cylinder Block
Sand cast in Hiduminium
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For all sand castings

experience has shown . . .

Hiduminium

makes the most of *Aluminium*

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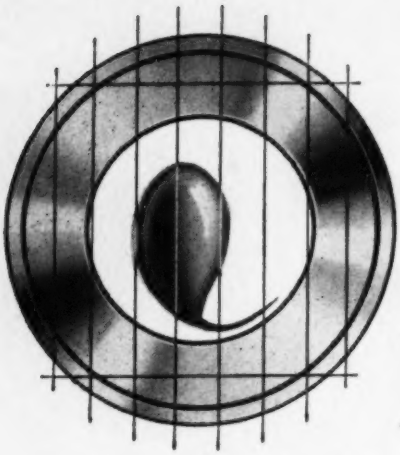
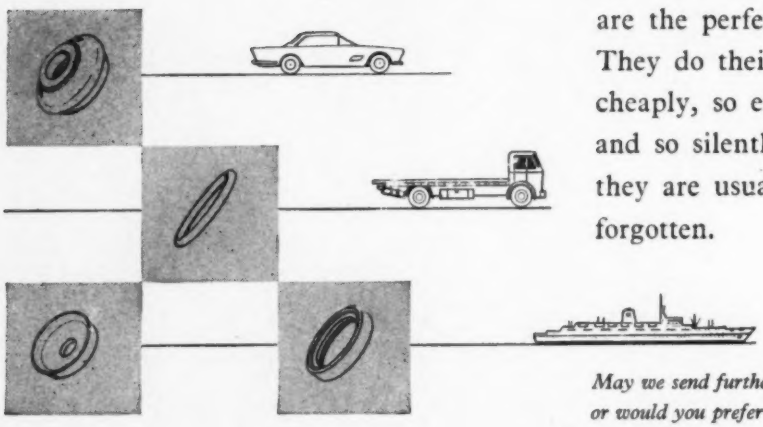


Oil

is a difficult prisoner...

and it must be kept in solitary confinement too; no fraternisation with that insidious enemy dirt, either. SuPerfect Oil Seals are the perfect **gaolers.** They do their job so cheaply, so efficiently, and so silently that they are usually completely forgotten.



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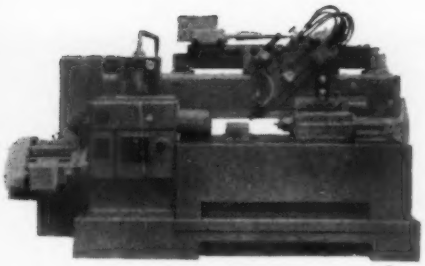
Manufacturers of 'SuPerfect' Hydraulic Packings and 'O' Rings; 'Romet' water pump seals and mechanical pump seals; 'Aeroquip' Flexible Hose with detachable re-usable fittings; 'Fidrac' mechanical rubber mouldings; 'Redcaps' Polythene Protective Caps and Plugs.



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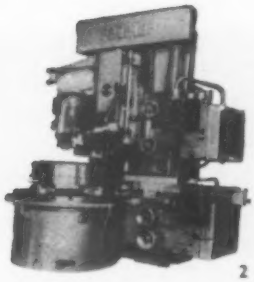

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When planning new production or considering replacement machinery contact Charles Churchill. Charles Churchill & Co. Ltd., are the selling organisation for a number of leading companies, British, Continental and American. The machines produced by these companies represent a comprehensive range of the finest machine tools available.

The Churchill list includes such famous names as

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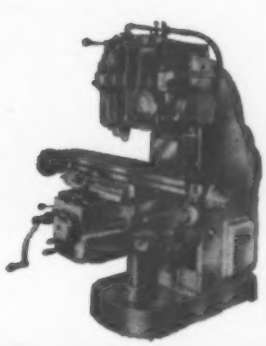
CONE AUTOMATIC MACHINE COMPANY LIMITED
S. A. JOS. PETERMANN
SCHARMANN & COMPANY
FRORIEP G.M.B.H.
ERNEST GROB
NATIONAL BROACH AND MACHINE COMPANY

There are Churchill branches at London, Glasgow, Newcastle, Manchester.

CHARLES CHURCHILL AND COMPANY LIMITED, COVENTRY ROAD, SOUTH YARDLEY, BIRMINGHAM 25

4



5

1 CHURCHILL REDMAN P.5 Automatic Multi-Tool and Profile Turning Lathe. One of the large number of lathes made by this company.

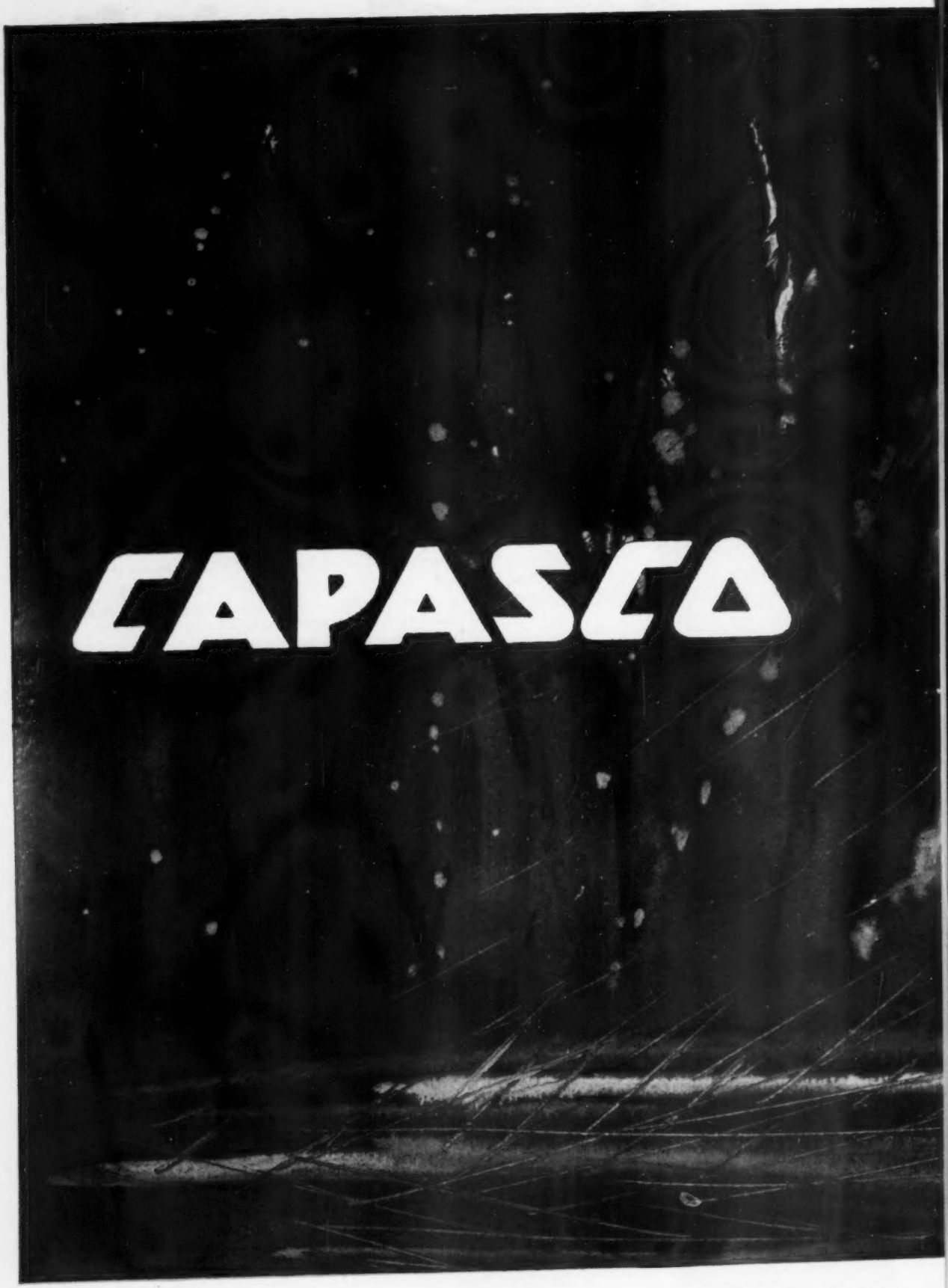
2 FRORIEP Vertical Turret Lathe with centralised remote control.

3 GROB. One of the many special purpose tools made by this company who also specialise in Thread Rolling Machines.

4 PETERMANN P.7 Automatic Lathe, representing the range of these fine Swiss-made machines.

5 CINCINNATI Dial Type Milling Machine.

CC.32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Summer Storms

Capasco takes care of the braking

BRAKE
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AUTOMOBILE PARTS

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We are specialists in the manufacture of precision and standard fasteners for the Motor Industry

NEWALLOY - Wheel Bolts, Wheel Studs.

NEWALLASTIC - Cylinder Head Studs and Bolts, Connecting Rod Bolts, etc.

NEWALLEX - Induction hardened Spring Shackle Pins and Bolts.

Standard bolts and nuts supplied in "R", "T", and "V" quality, and other Alloy Steels.

Group of typical components, as supplied to the leading motor car manufacturers.

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MAXICUT

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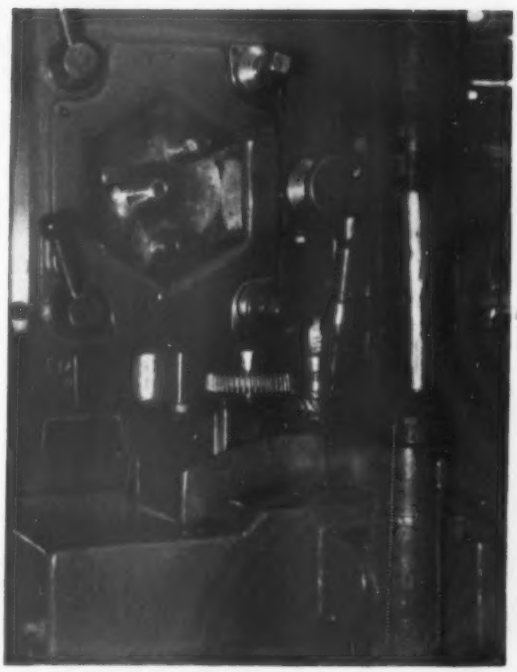
No. 2A
Gear Shaping Machines
with Automatic loading

These Drummond Gear Shapers can now be supplied with automatic loading equipment for the mass production of specific types of gears. When operated in conjunction with conveyor systems, gear production becomes a completely automatic process.

The illustration above shows a line of Maxicut machines installed for high output production of automobile gears and in the close-up view gear blanks can be seen stacked in the tubular type magazine. From this magazine a slider type feeder loads the blank on the work spindle and displaces the machined component into an exit chute. A push button panel at the left of the machine incorporates switch controls for hand or automatic operation, also controls to repeat or not-to-repeat the cycle. The indicator light panel on top of the machine shows whether all functions are operating correctly or not.

Write to-day for full details of Drummond Maxicut Gear Shapers for high speed accurate gear production.



This view shows the slider type feeder arrangement and magazine containing gear blanks.

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the spray gun for your industry

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MAGNETOS.....

but frankly, Simms
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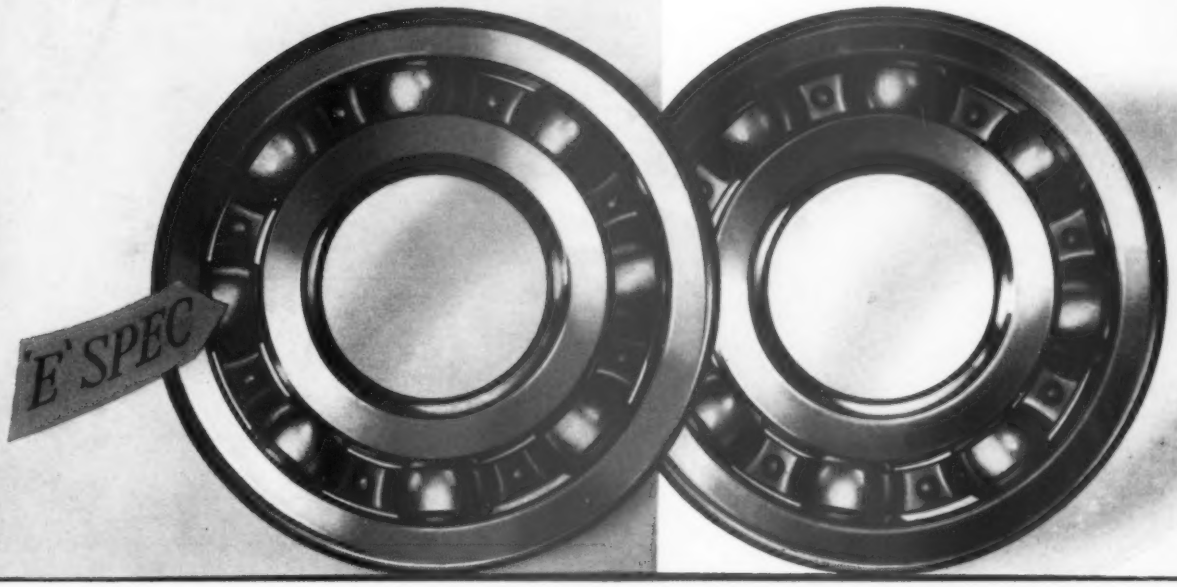

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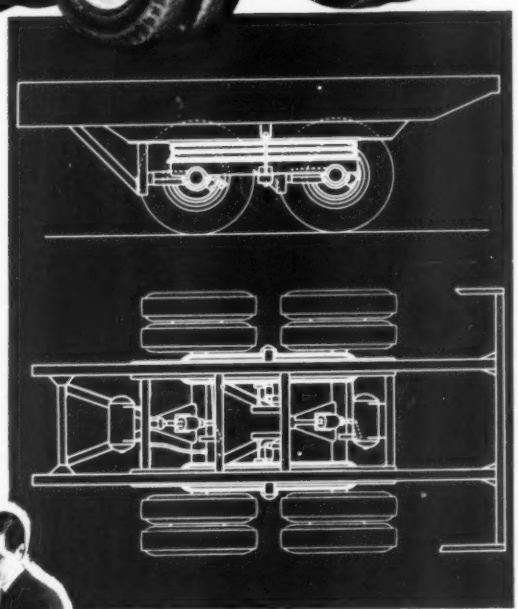

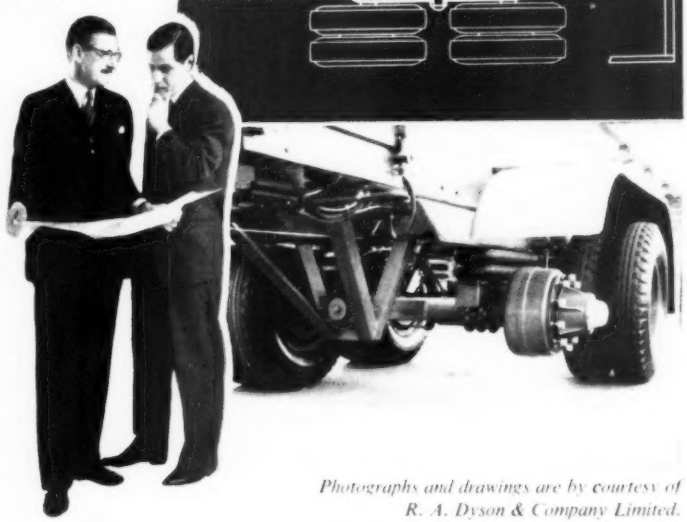

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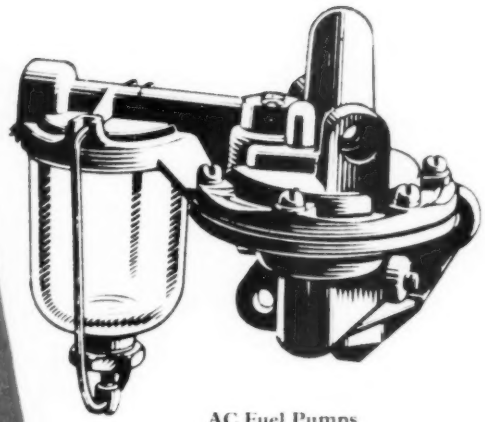
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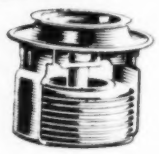
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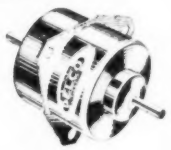
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a good vehicle starts with

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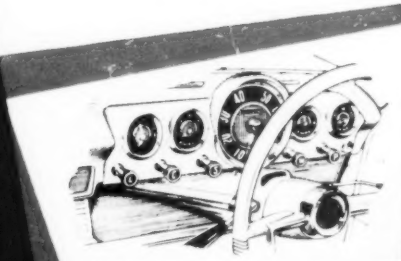
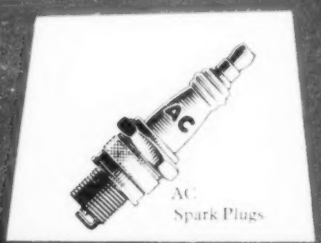

AC Instrument Panels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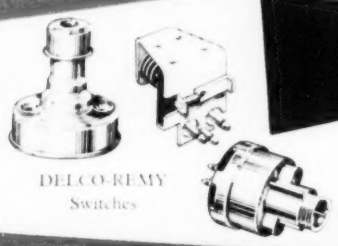
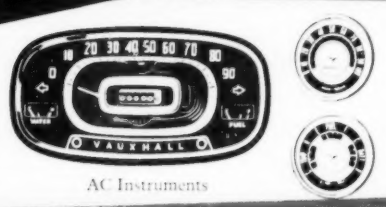

DELCO-REMY Switches



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The illustration shows one of our electronic measuring machines specially built for us and installed in the Piston Inspection Line at our Warwick Factory. The visual indicator has been enlarged to show in detail the arrangement by which seven dimensions are accurately checked in one operation, speeding production and ensuring consistency of measurement without risk of human error.

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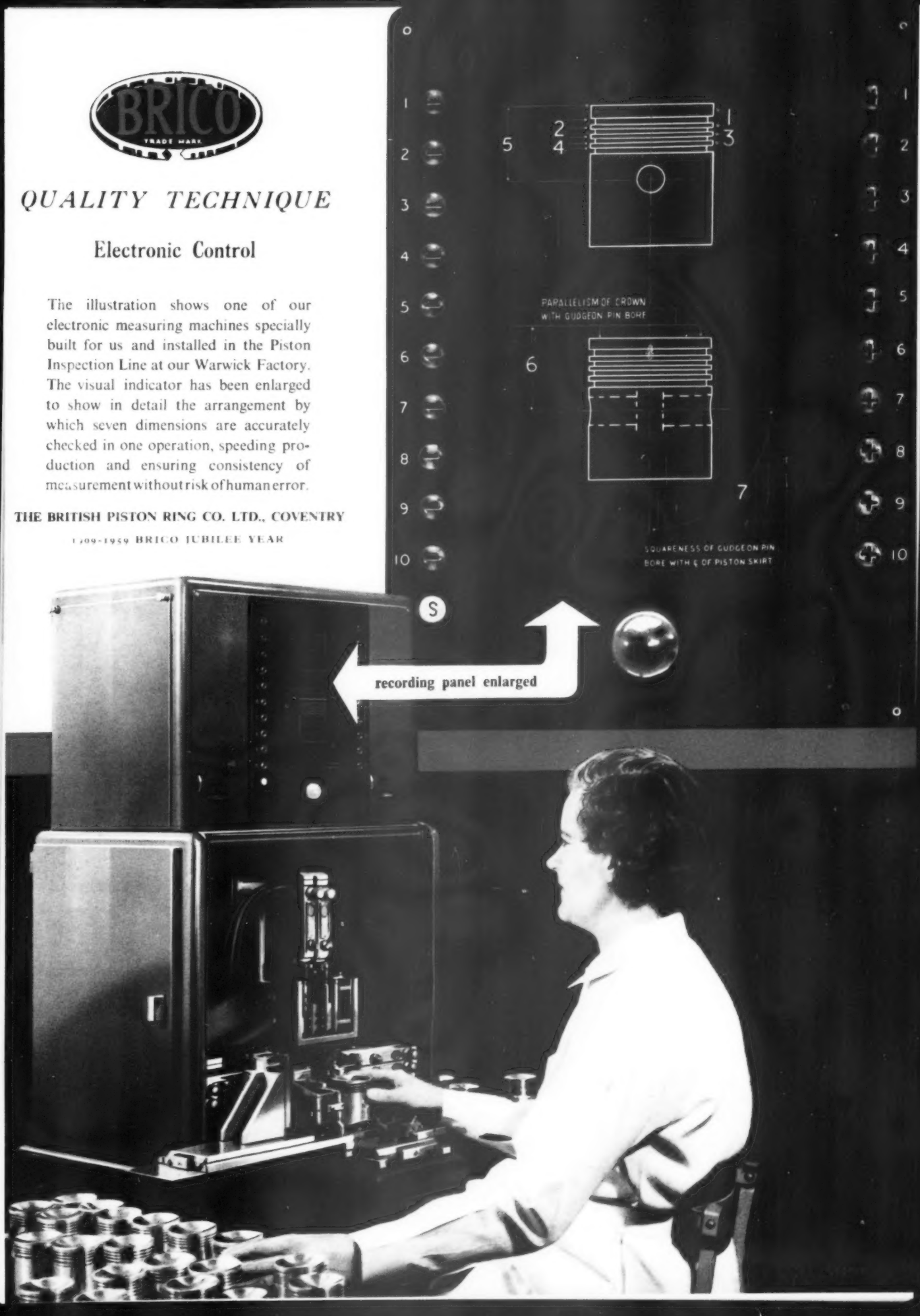
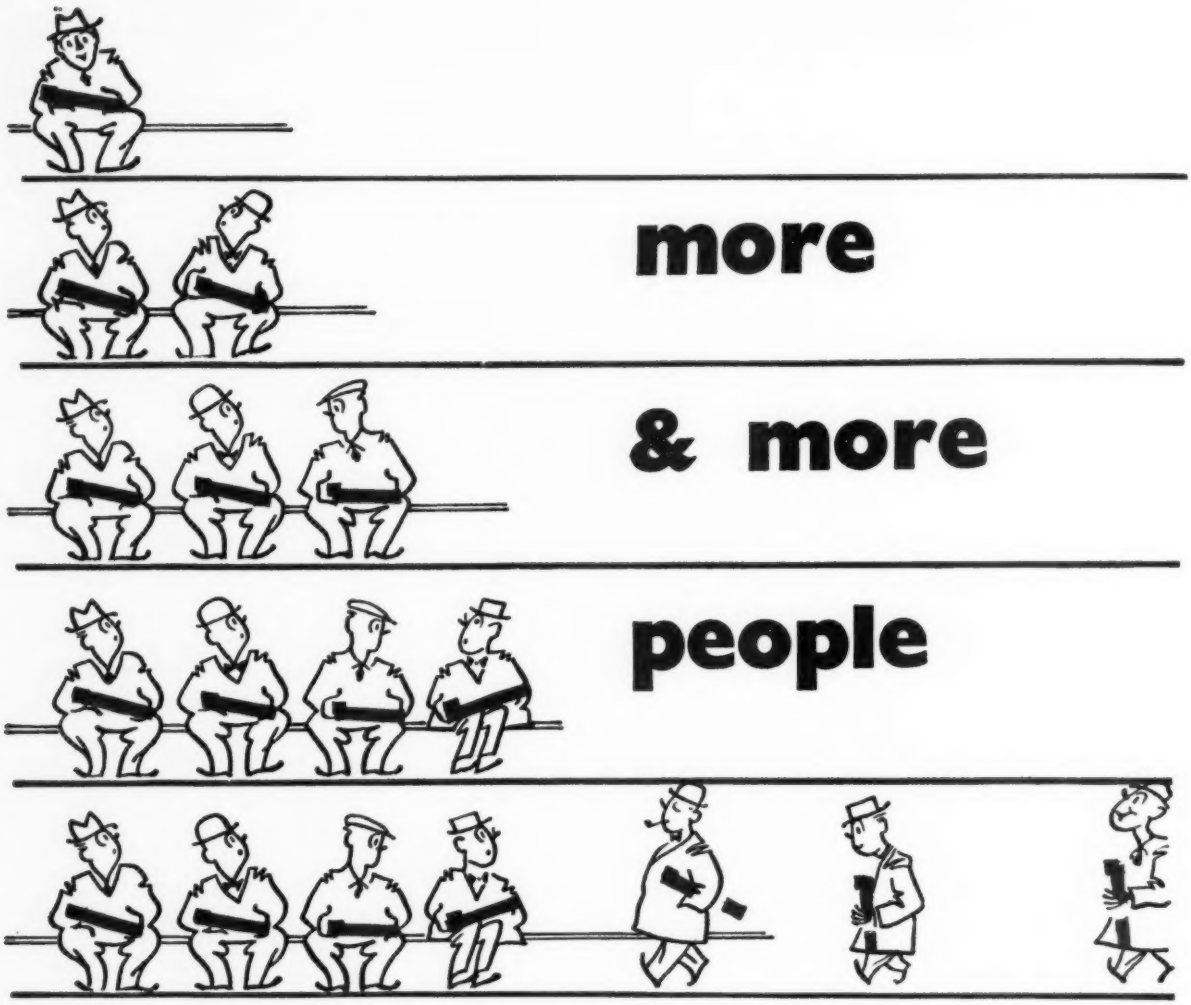
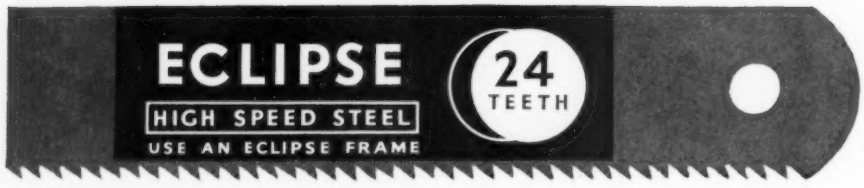

PARALLELISM OF CROWN WITH GUDGEON PIN BORE
SQUARENESS OF GUDGEON PIN BORE WITH ℄ OF PISTON SKIRT
recording panel enlarged



use Eclipse



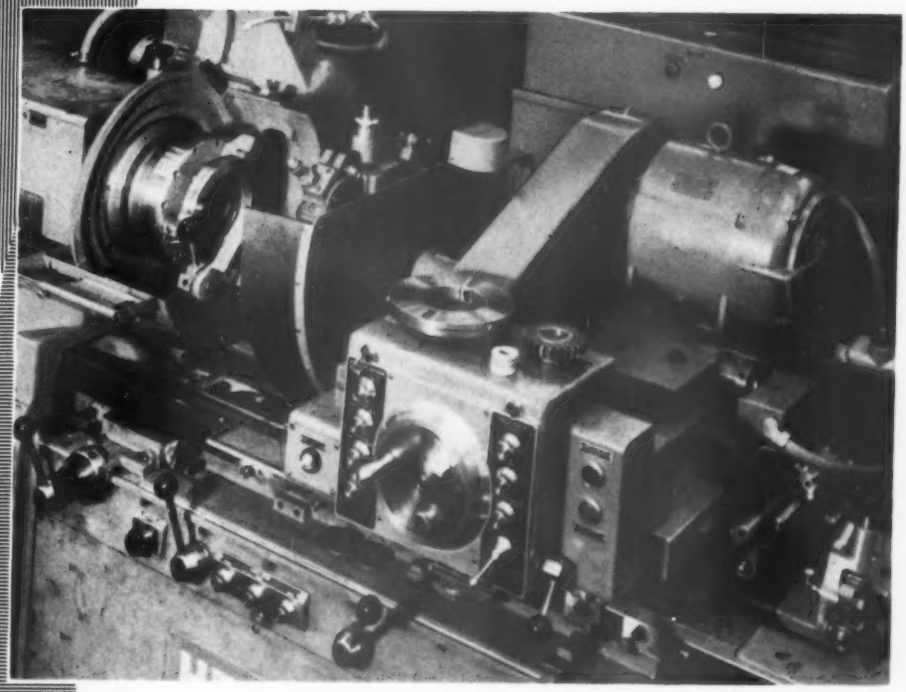
'Eclipse' hacksaw blades and other tools are made by James Neill & Co. (Sheffield) Ltd. and are obtainable from all tool distributors.

UH 35

cutting costs on a

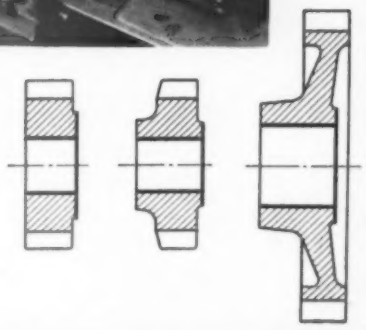
HEALD INTERNAL GRINDER

precision grinding four different gears in one set up on one machine



These gears are of three different contours as shown. Finish in the bore is 15 micro-inch and on the face 20-25 micro-inch. Bore and face are square to .0005". Stock removal .012".

Gear is located in an adapter and held in a pneumatic diaphragm chuck. Bore is ground with table reciprocation and standard Sizematic cycle. When finished to size wheelhead is indexed forward and face is plunge ground with the facing attachment.



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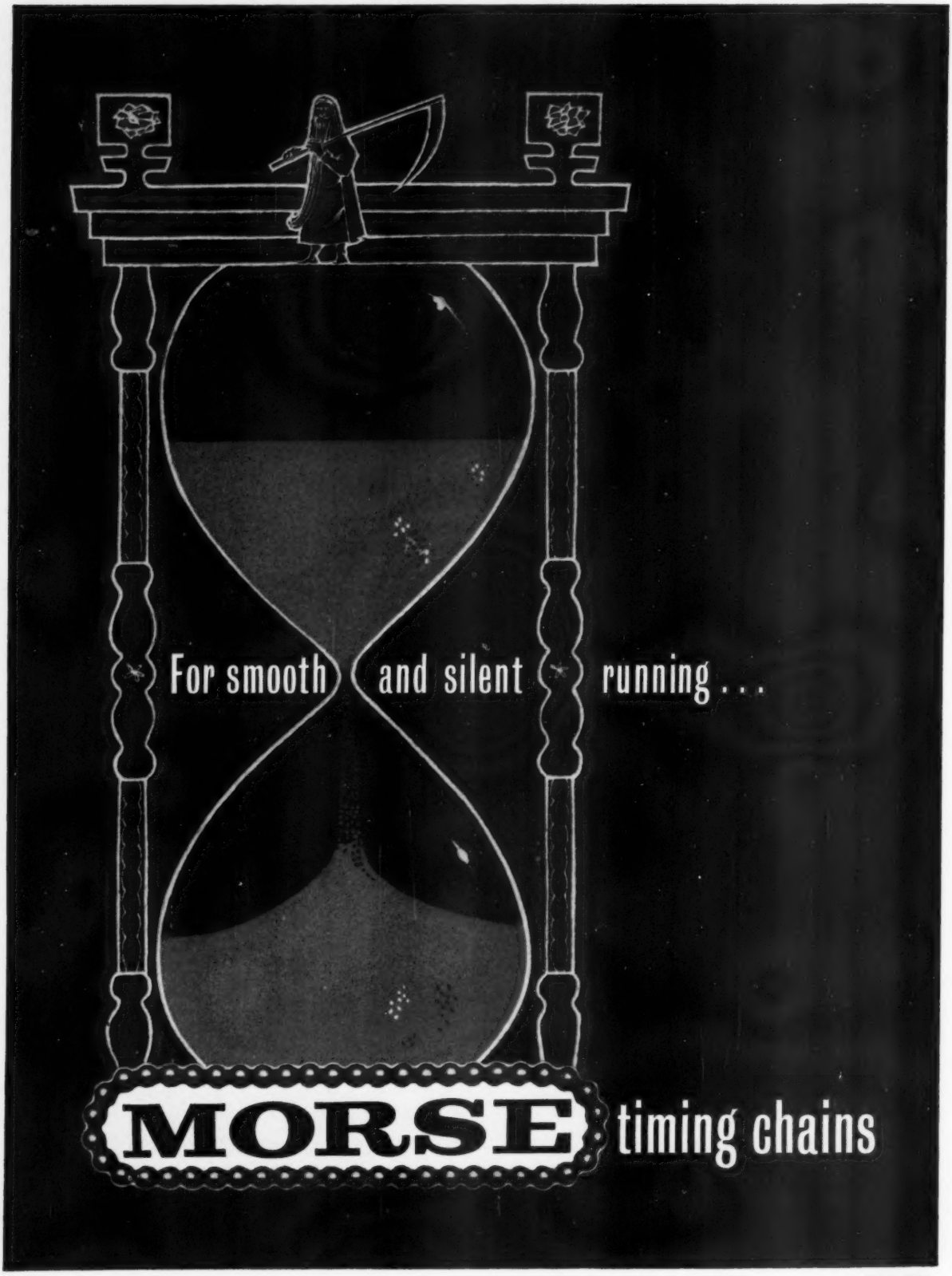

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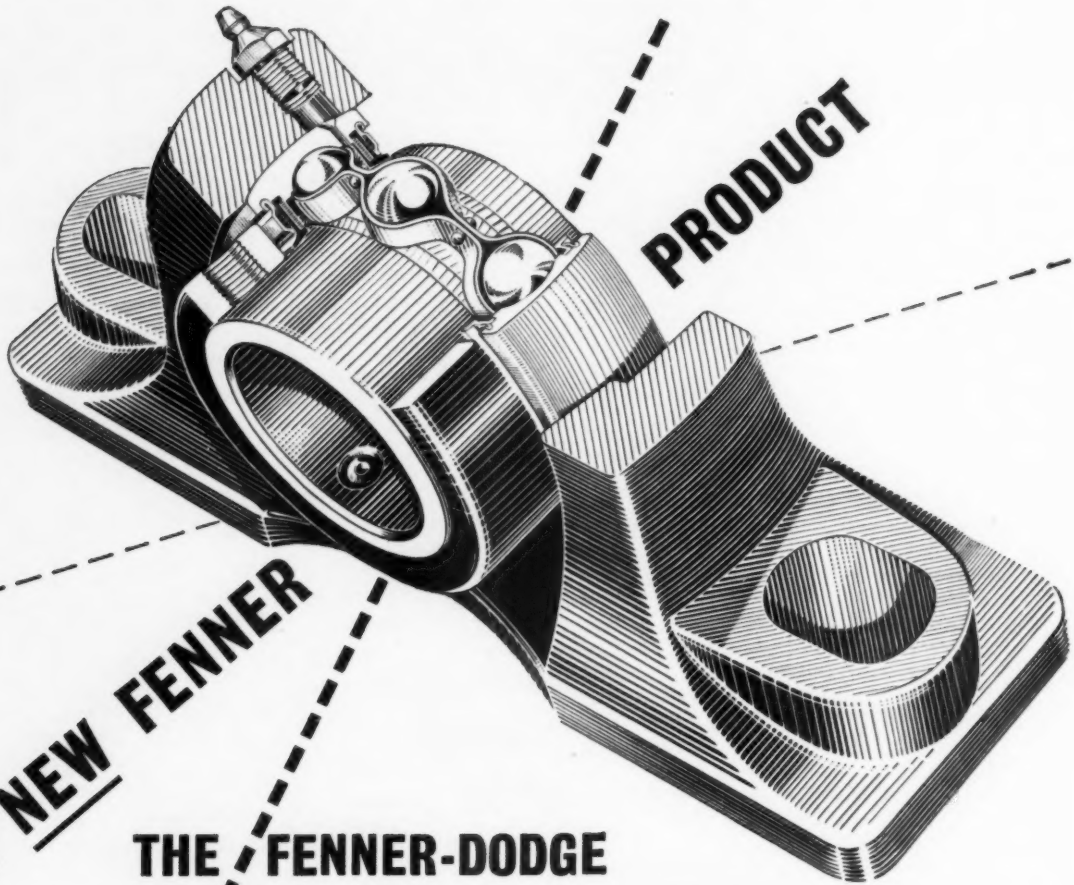

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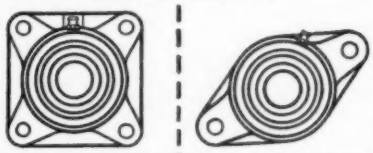

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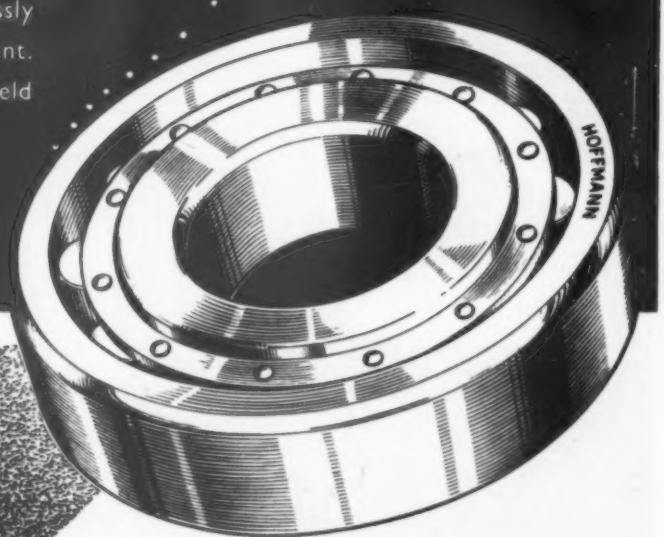

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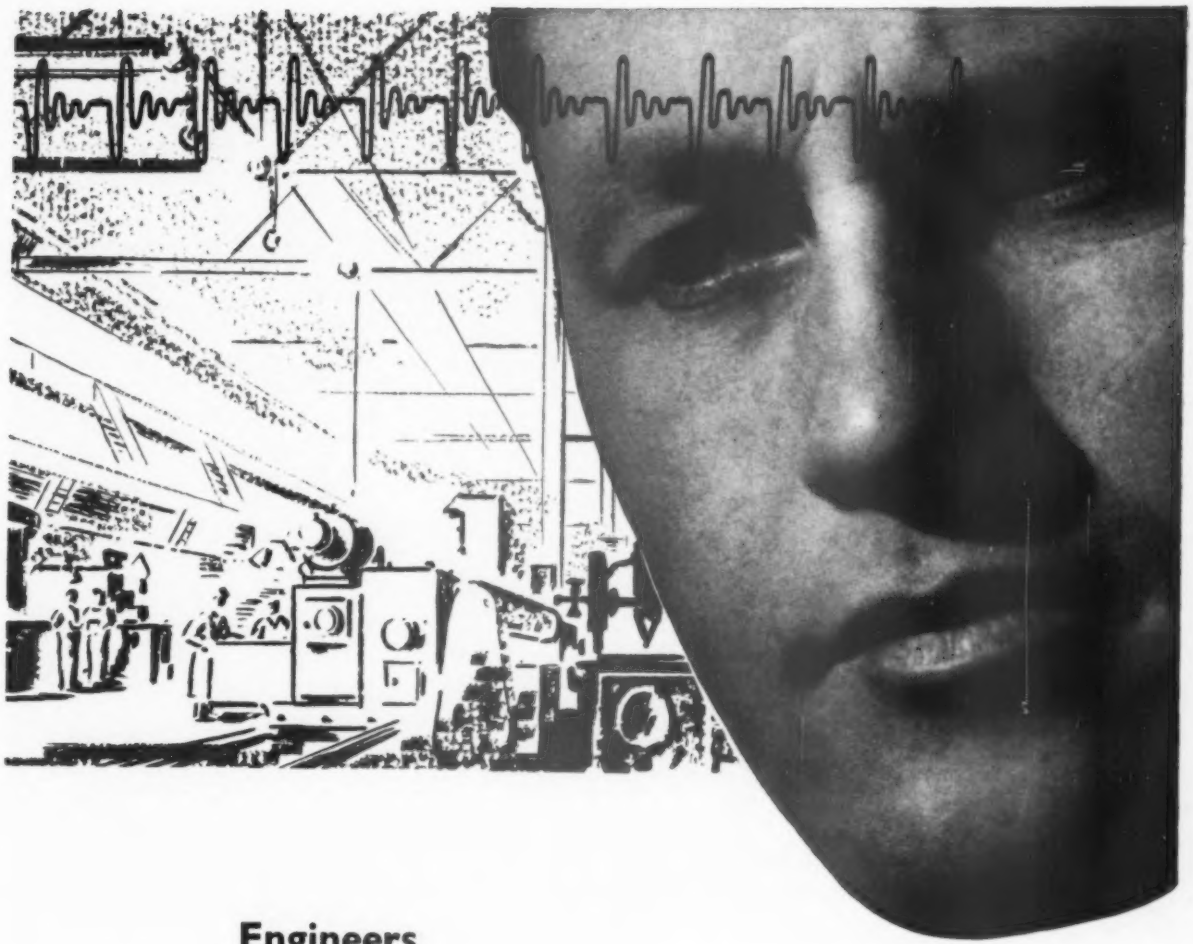

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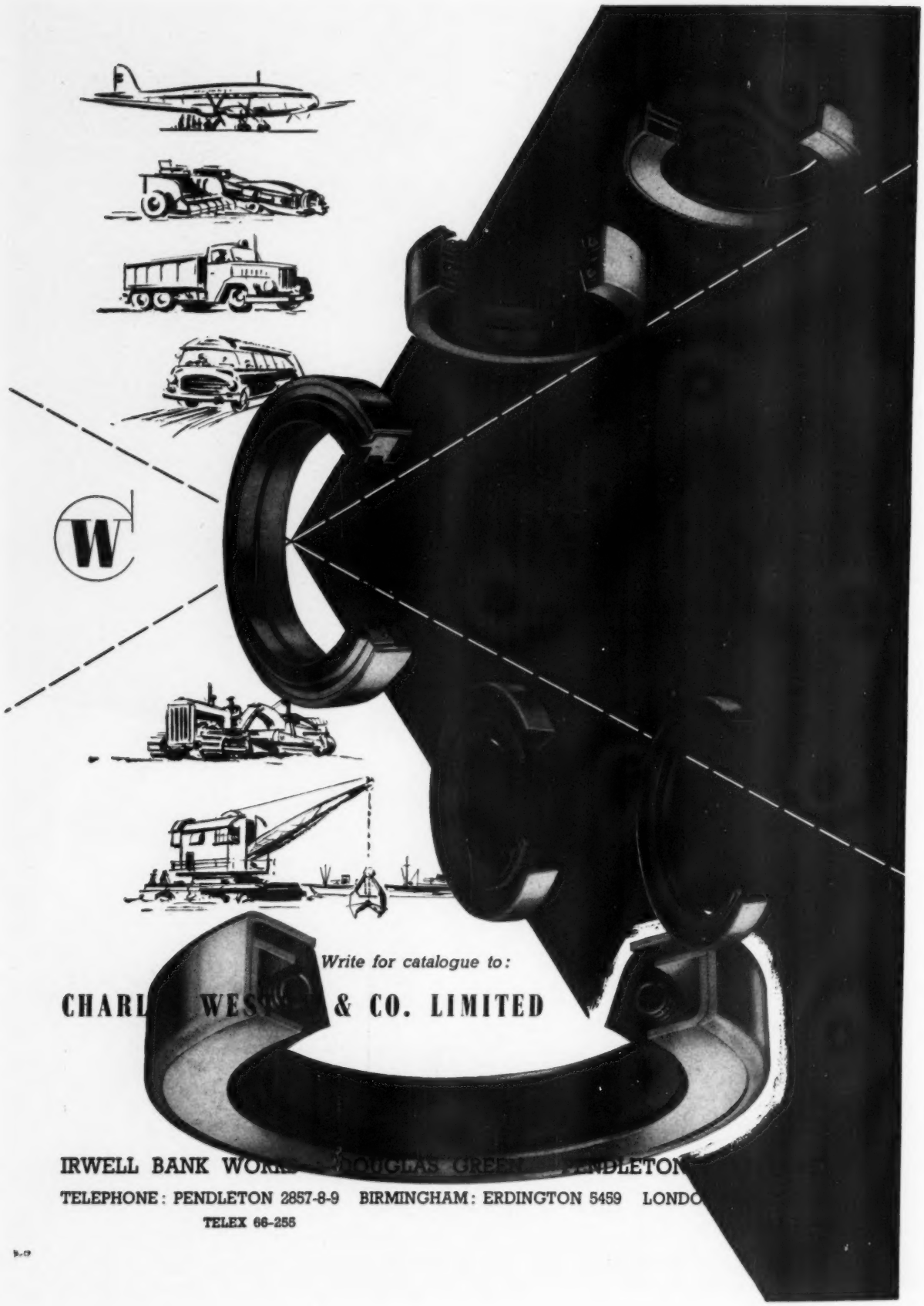
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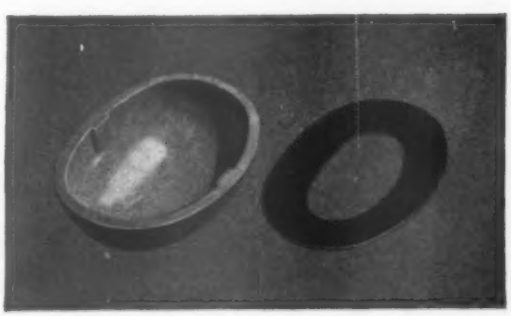
Here's a typical example of how the mechanical strength of Beetle D.M.C. (Dough Moulding Compound) helps to solve industrial problems.

These ear defenders are used to protect the hearing of workers in proximity to jet engines and in noisy factories. During the development work by the makers, Dennis Ferranti Meters Ltd., twelve sets moulded from conventional materials were delivered to Bomber Command for testing. The report of The Central Medical Establishment, Acoustics Laboratory, was that the ear defenders gave excellent protection, but that the mouldings then used were considered too fragile.

D.M.C. mouldings were substituted and have been found virtually indestructible.

D.M.C. is the ideal material whenever high mechanical and/or electrical strength is required. The material can be moulded in conventional compression presses on short cure times and at low pressure. We shall be glad to send full technical details.



This illustration shows the main moulding from Beetle D.M.C. and a retaining piece made from Scarab material. The Earguards are used in engine testing, rivetting, blasting, rolling mills, weaving sheds, paper mills and on aerodromes.

B·I·P

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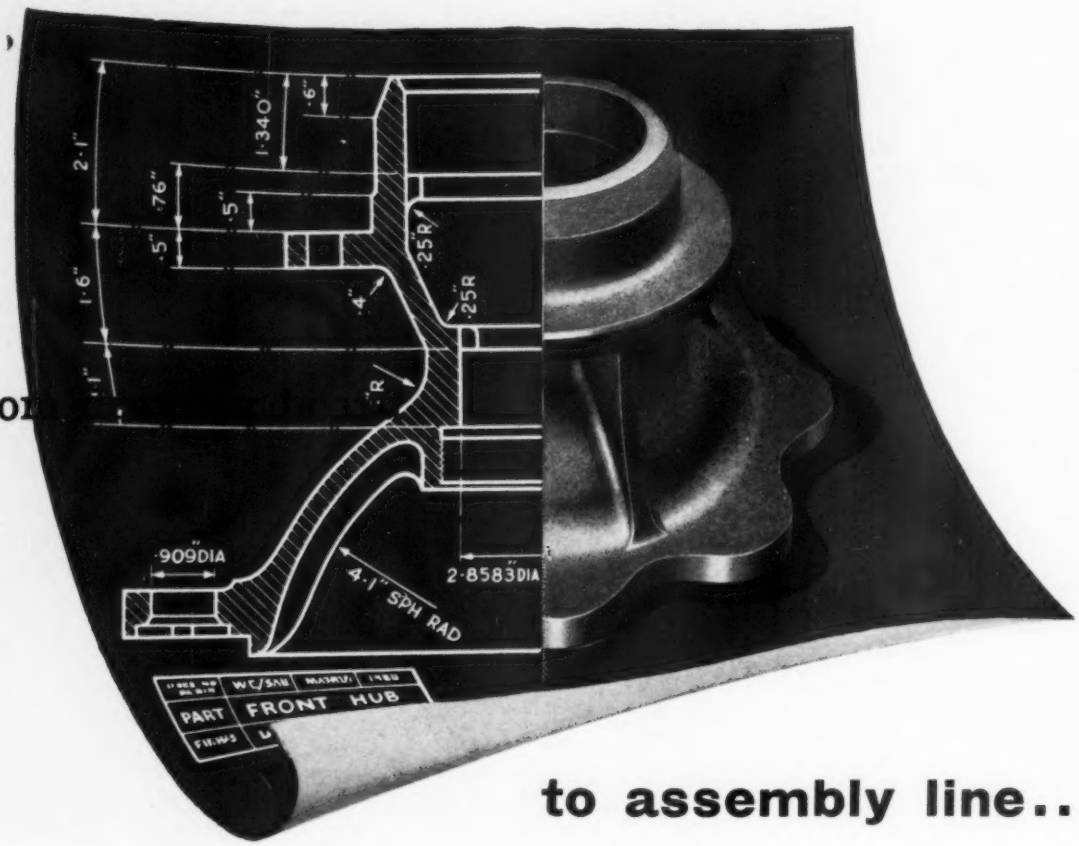

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to assembly line...

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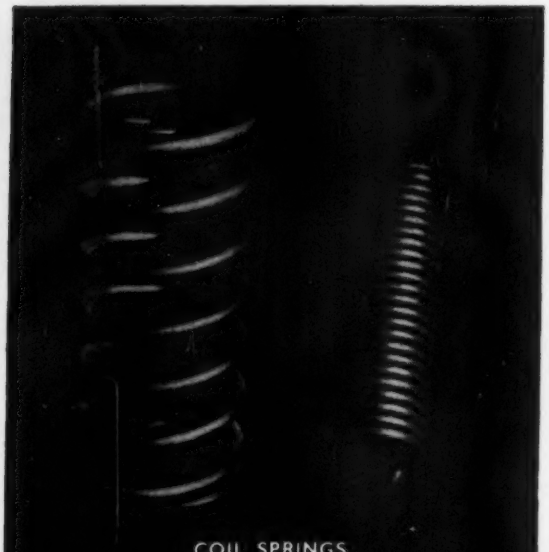

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give top performance everywhere

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also
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We are also the manufacturers of

KANTLINK
SPRING WASHERS

LAMINATED SPRINGS

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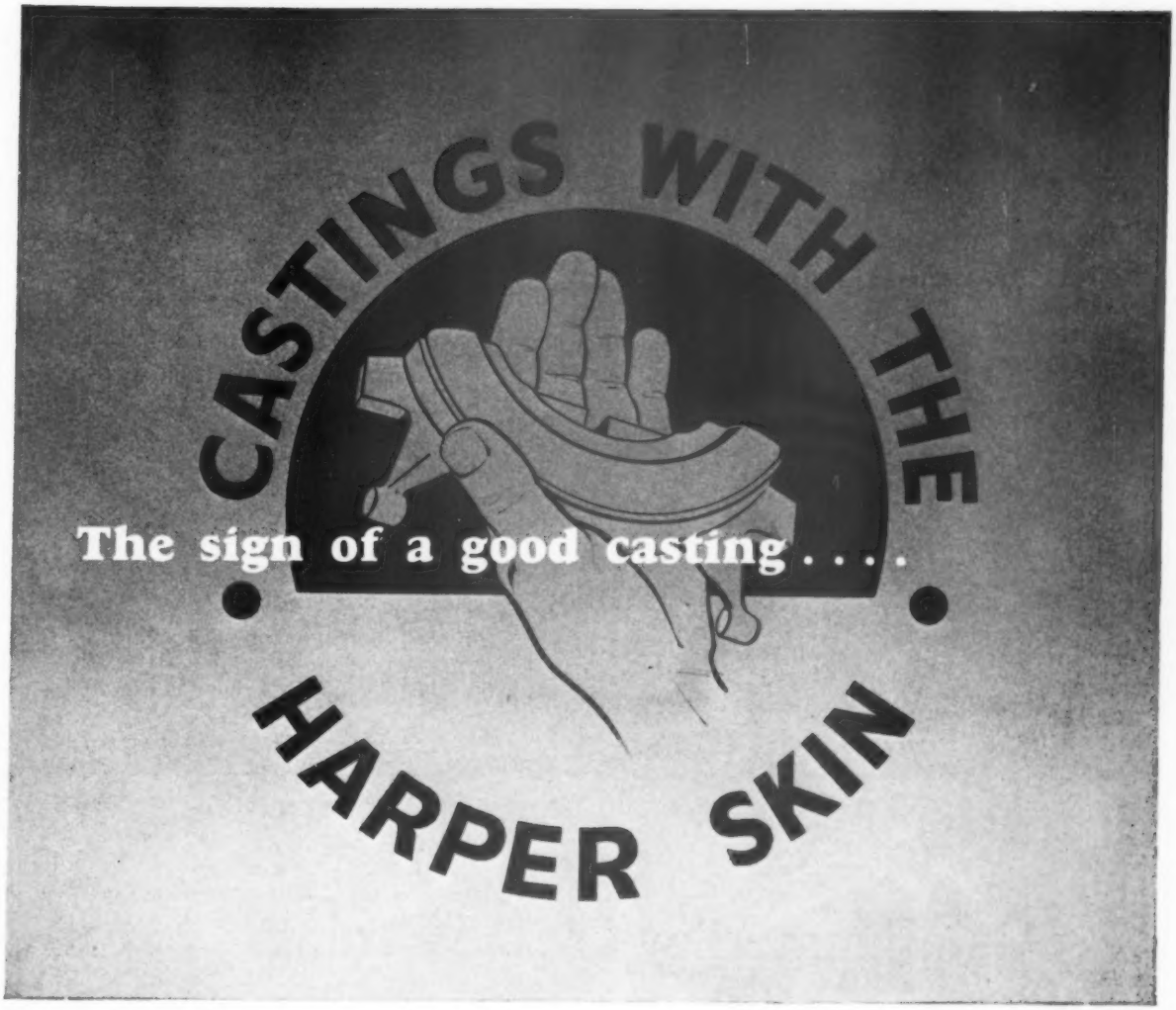

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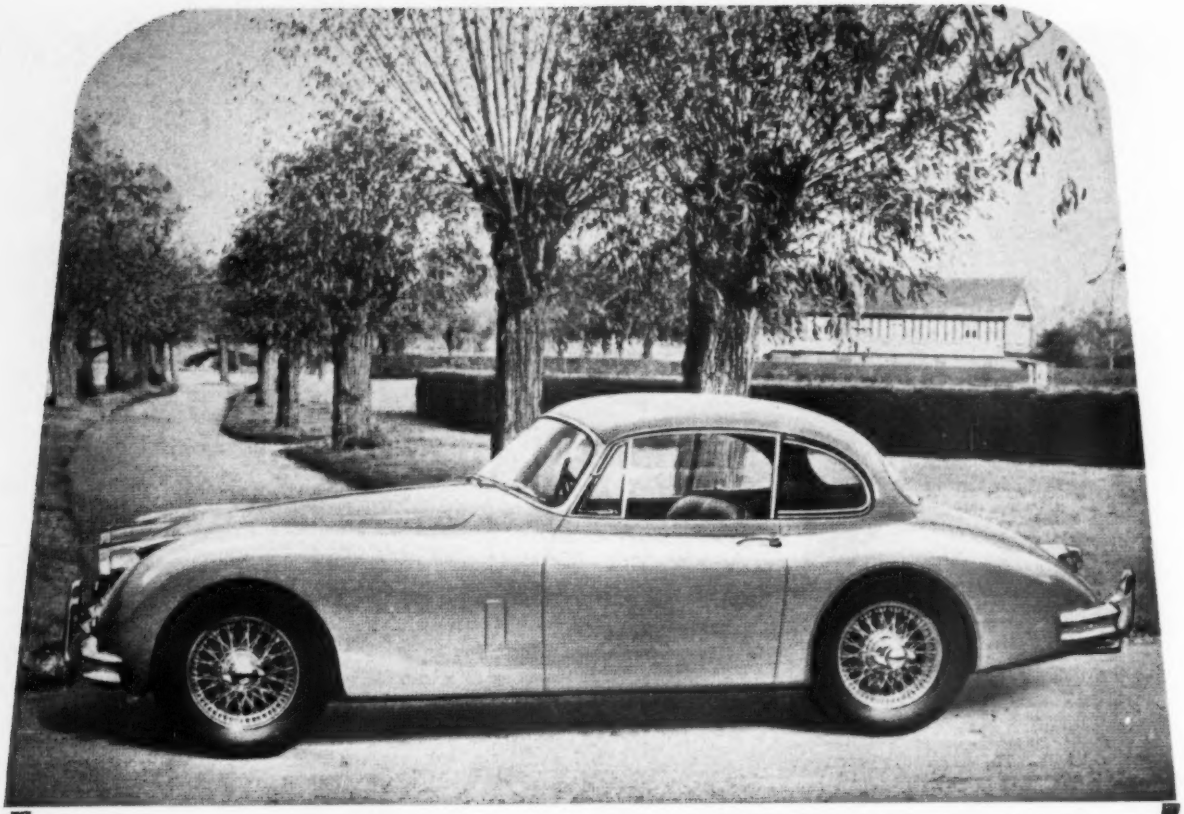
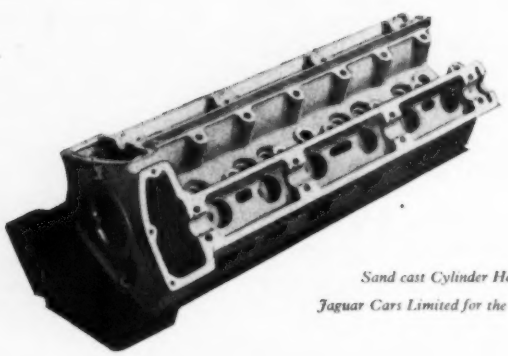


Photograph of the new XK150 by courtesy of Messrs. Jaguar Cars Limited

Aluminium Alloy Engine Castings

by William Mills Limited



Sand cast Cylinder Head as regularly supplied to Jaguar Cars Limited for the above model

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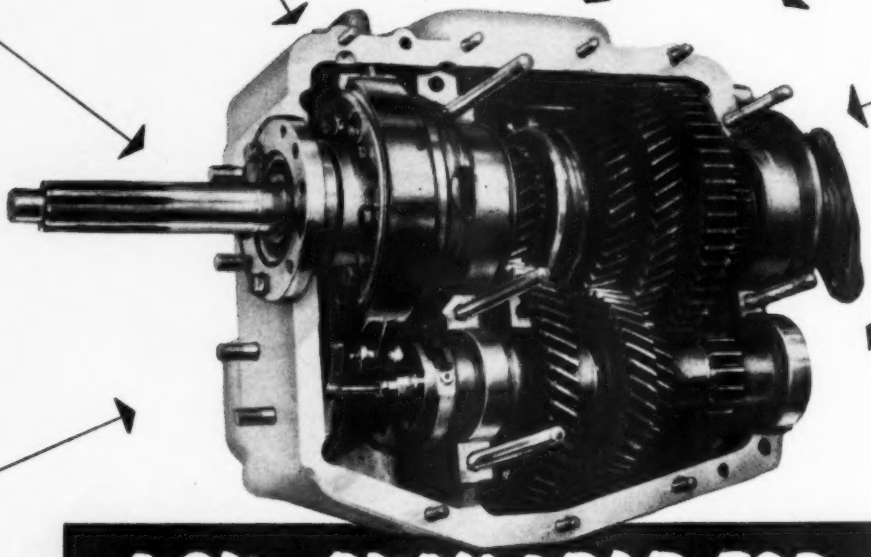

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oil-cooled
WELDING TRANSFORMER

for heavy outdoor all-weather work

★ **TWIN CURRENT RANGE**

HIGH range for plate welding . . 65 V at 50—310 A
LOW range for sheet metal work . 95 V at 40—220 A
Rated to BSS 638/1953 for continuous duty at maximum current . Load 20 kVA

★ **INFINITELY VARIABLE CURRENT CONTROL**

Over the full range. Current setting by handwheel. Indication from easily read scales mounted on machine top.

★ **HIGH POWER FACTOR**

The power factor is within the recommendations of the B.E.A.M.A. A built-in capacitor can be supplied if required.

★ **ROBUST CONSTRUCTION**

Heavily built oil tank, hermetically sealed to enclose the vacuum impregnated coils.

★ **LIGHT AND MOBILE**

Mounted on four wheels, the leading pair swivelling for easy manoeuvrability.

ELECTRODES
A wide range of Actarc electrodes is available. There are lime-ferritic electrodes for welding medium carbon and low alloyed steel; mild steel electrodes for general purpose work; special rods for vertical and overhead welding, and for deep penetration; electrodes for fast horizontal production work; electrodes for boiler plate; non-ferrous electrodes for aluminium, nickel and bronze.

Range of ACTARC Transformers.

Air and Oil cooled MEDIA transformers ...	Low Voltage range 65 V	50-310 A
	High Voltage range 95 V	40-220 A
Oil cooled MASTA transformer	Low Voltage range 65 V	70-440 A
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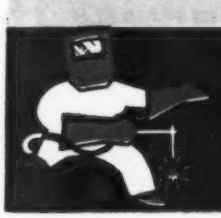
Air cooled MONTA transformer	Low Voltage range 50 V	30-200 A
	High Voltage range 90 V	15-100 A
Two easily selected infinitely variable current ranges.		
Oil cooled MODULA (Tapped Choke) ...	Low Voltage range 50 V	30-250 A
	High Voltage range 80 V	40-185 A

Suitable for electrodes of the lime-ferritic and deep penetrating types and other special electrodes requiring high striking voltages.
All transformers can be supplied for standard voltage supplies for connection to single or 3-phase, 50 cycle A.C. systems.
Special models available for other voltages and frequencies.



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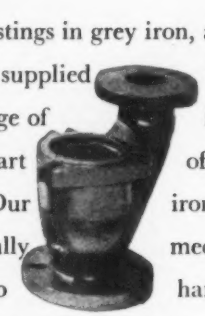
P 10

QUALITY CASTINGS

FOR QUANTITY SUPPLY....in Iron or Aluminium Alloy



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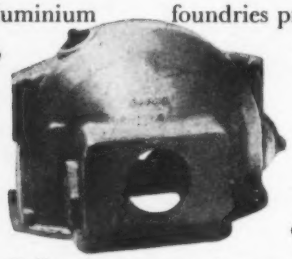

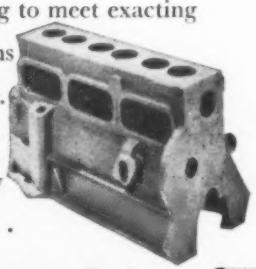

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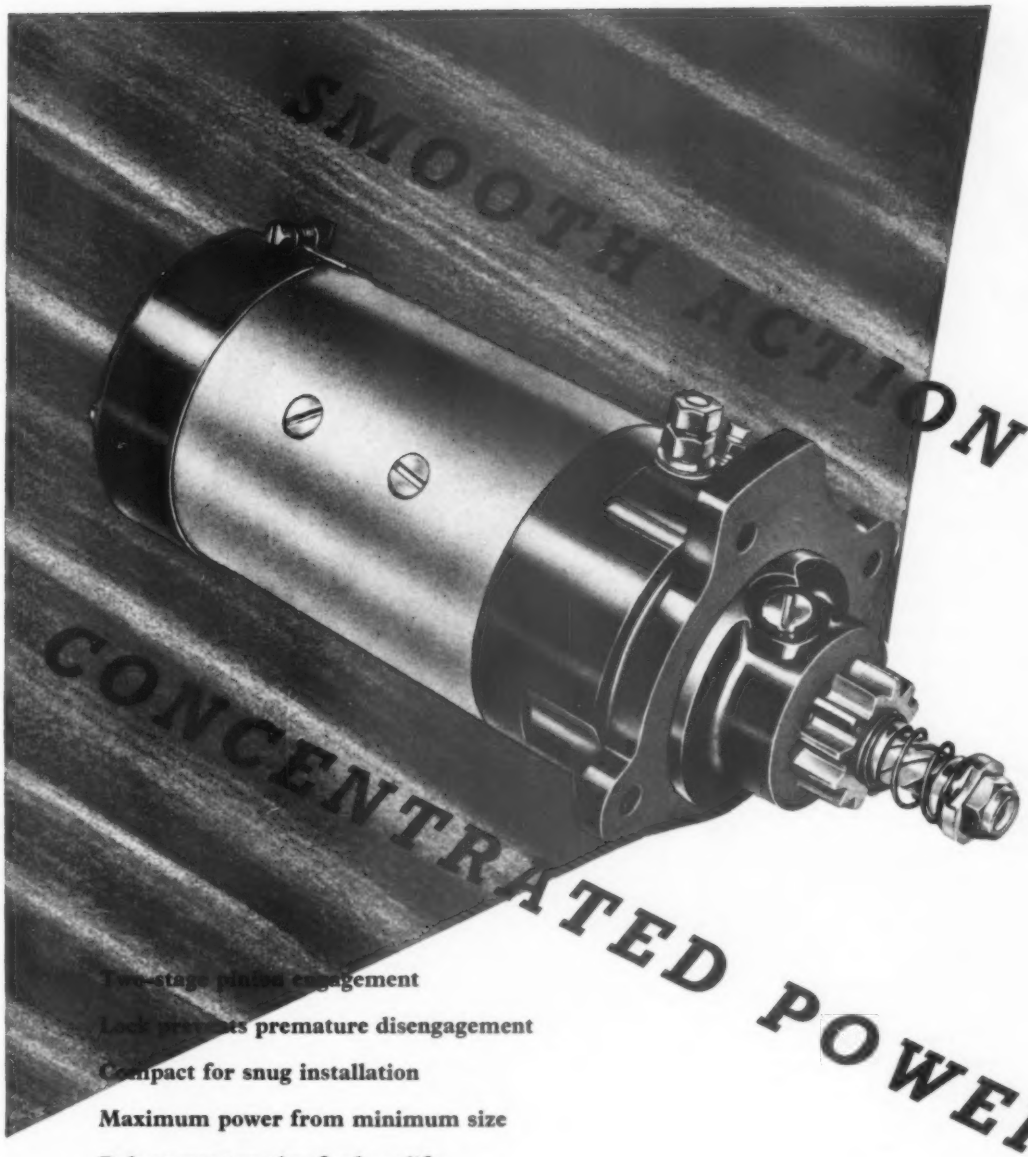

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Lock prevents premature disengagement

Compact for snug installation

Maximum power from minimum size

Robust construction for long life

In the CA45 starter a great deal of power is packed into a small unit. Although only 4½ inches in diameter, the CA45 is suitable for heavy duty and will crank multi-cylinder diesel engines of up to 5 litres capacity. The pinion on its sleeve alone moves axially, operating a trip mechanism which ensures that full torque is applied only when the pinion is in proper engagement. A locking device prevents premature disengagement of the pinion while the starter switch is depressed. Results of exhaustive tests, both in the laboratory and in the field, show that the CA45 is efficient, reliable, and has an exceptionally long life between overhauls.

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AP931

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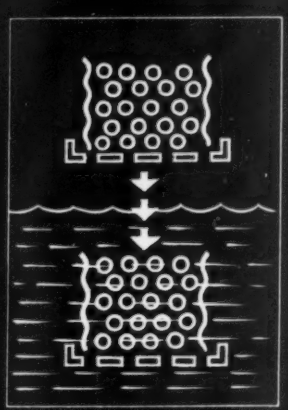

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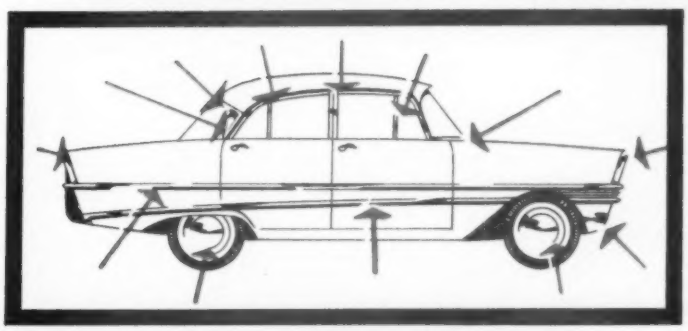

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Sparkling beauty enhances styling of modern cars
Cannot rust, peel or chip. No pitting.
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Bright right through.
Easy to clean (using only soap and water)
Time-proof brilliance keeps up value of car.
Already adopted for many parts both outside and in by leading manufacturers, such as Vauxhall, Morris, Wolseley, Riley, etc.

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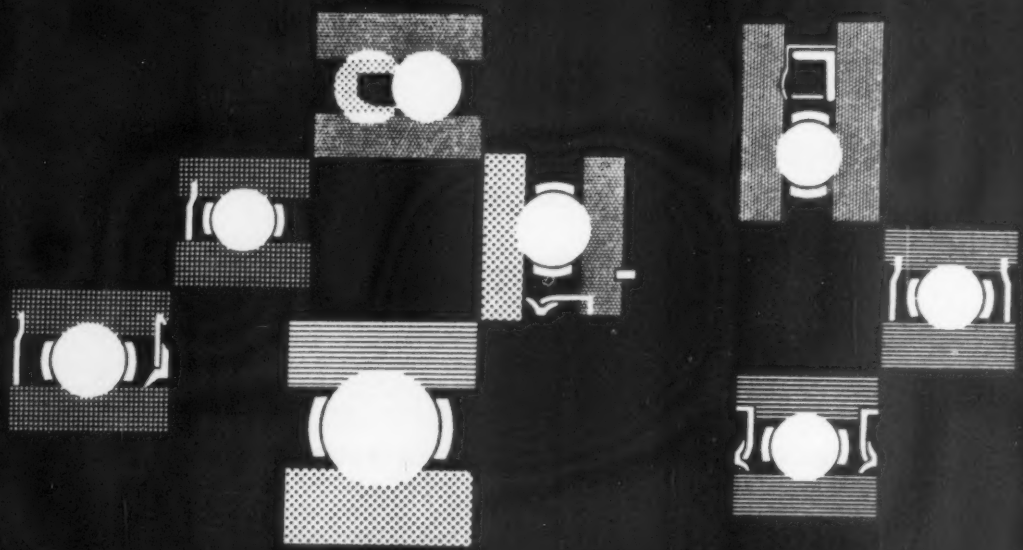

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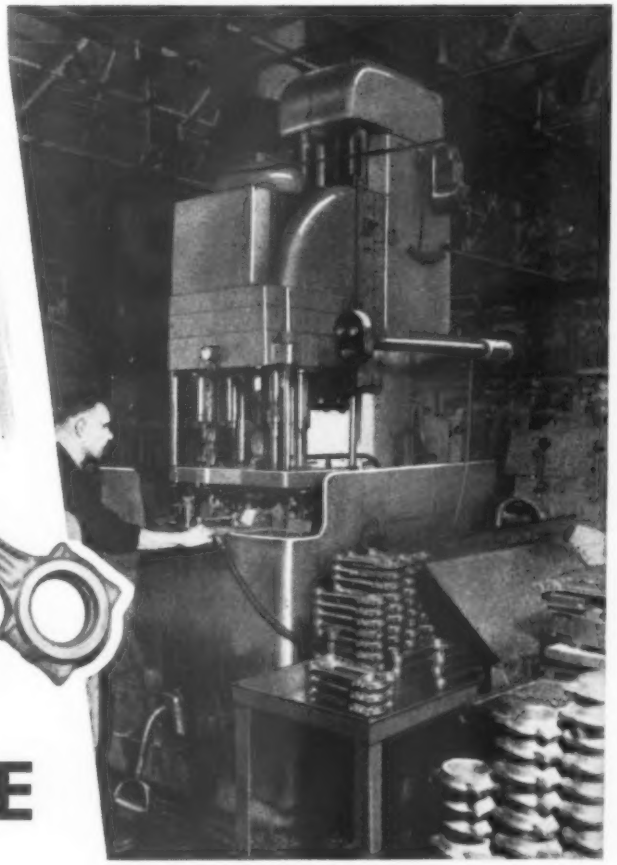

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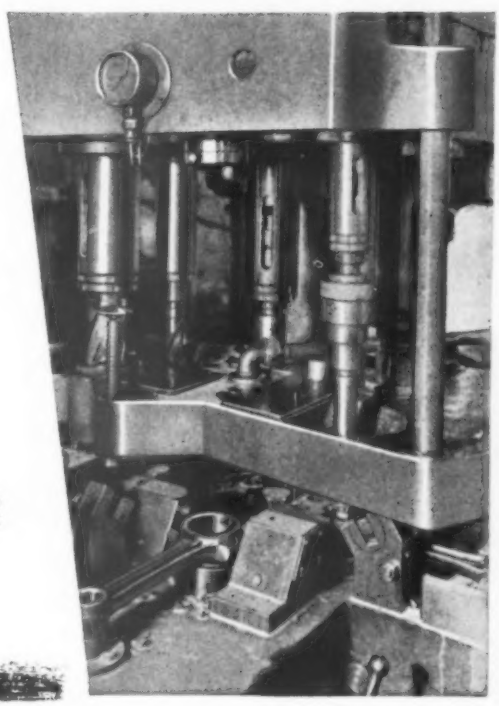
cuts costs on 'JAGUAR' connecting rods

At Jaguar Cars Ltd., Coventry, this Size 2 ARCHDALE hydraulic feed multi-drill is fully meeting the demands of expanding production schedules. Using a four-station indexing table, with one loading and three working positions, connecting rods are produced at fast rates. Small ends are drilled and reamed, and the large ends are core drilled and bored. Small end bore is $\frac{43}{64}$in. dia., and large end 2.200in.

If your problem calls for standard or special multi-drilling ARCHDALE have the answer. Ask for production estimates.



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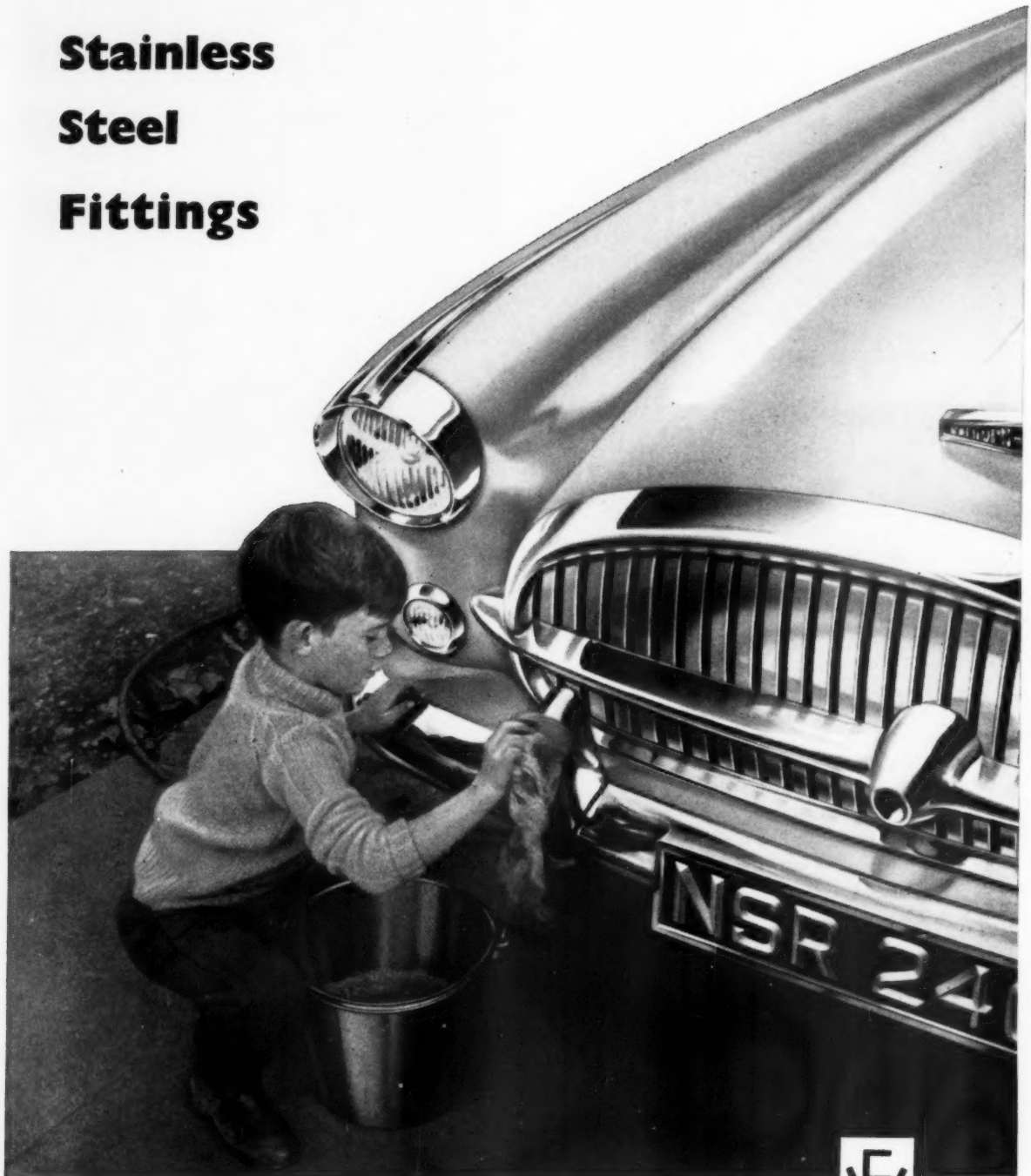

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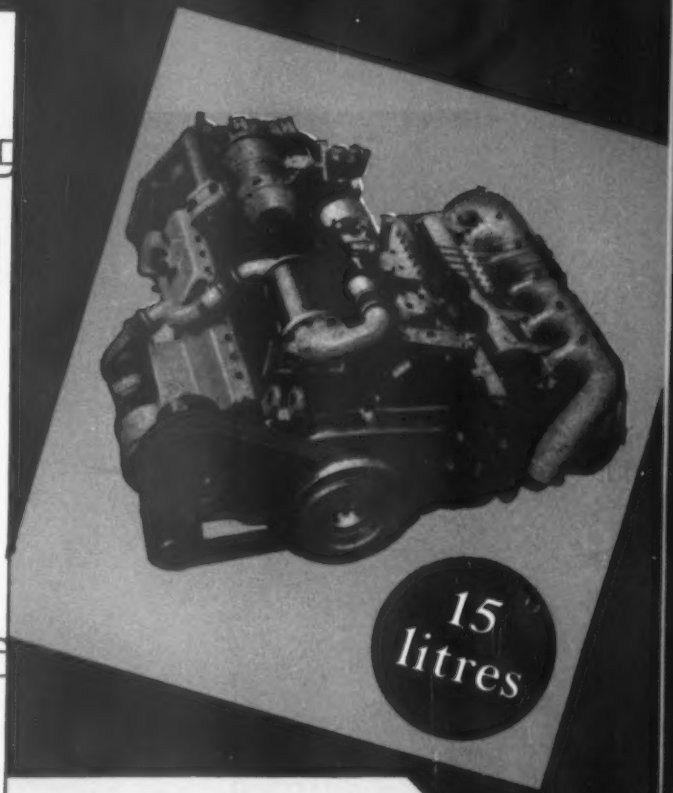

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Experience, essential in torsional vibration damping, ensures the success of Metalastik T.V. Dampers. For more than 20 years Metalastik has designed and manufactured dampers for world-famous engines.

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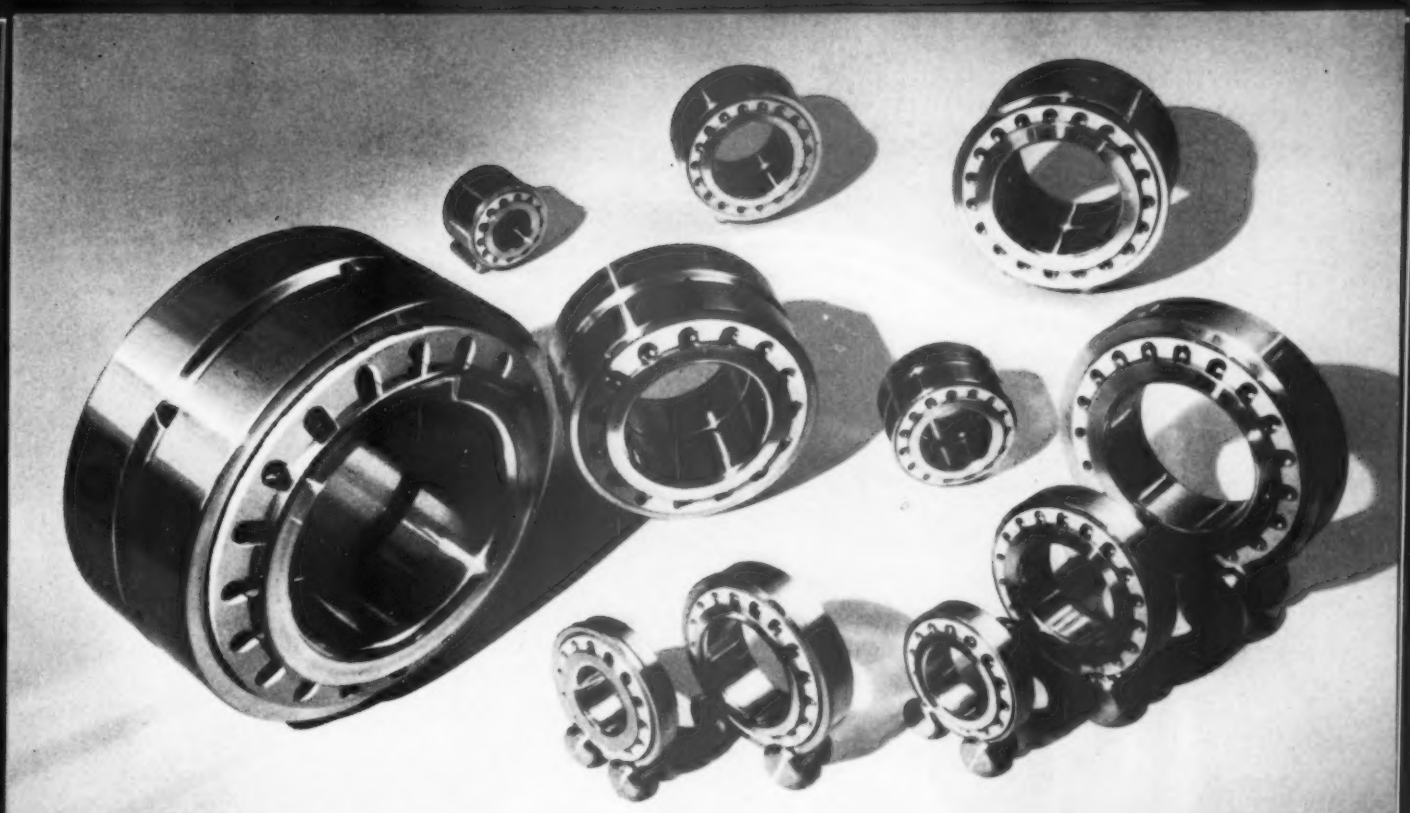

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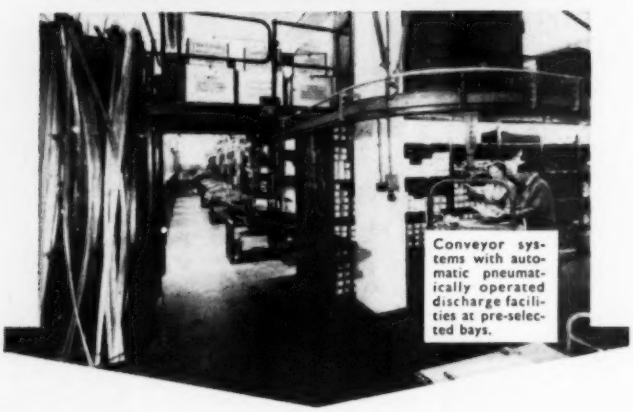
Enots staff operating the modern stock control system.

This photograph illustrates the modern pre-packaging bay.



Conveyor systems with automatic pneumatically operated discharge facilities at pre-selected bays.

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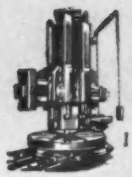
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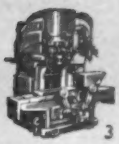
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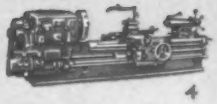
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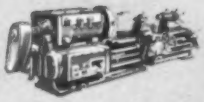
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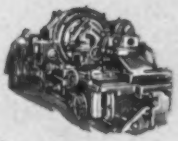
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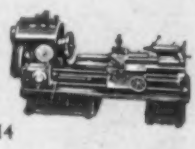
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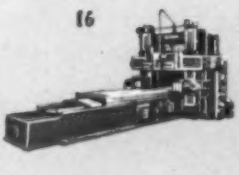
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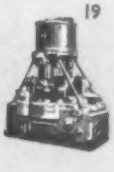
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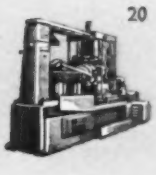
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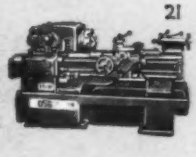
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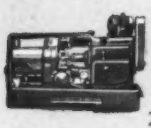
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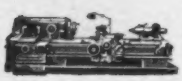
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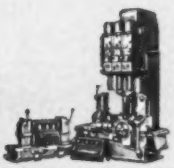
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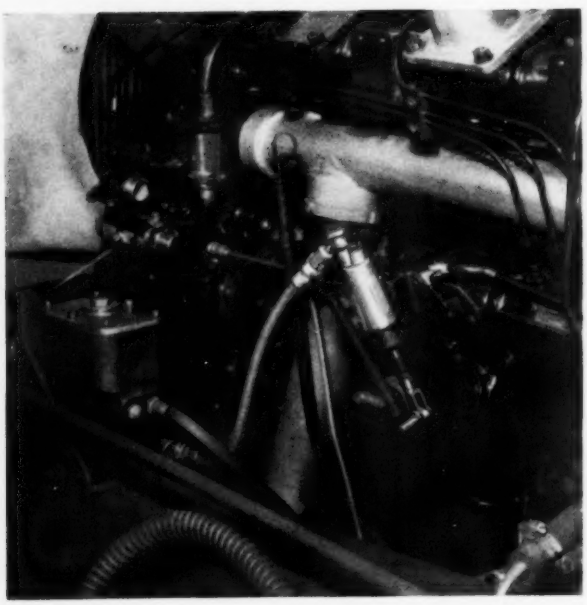

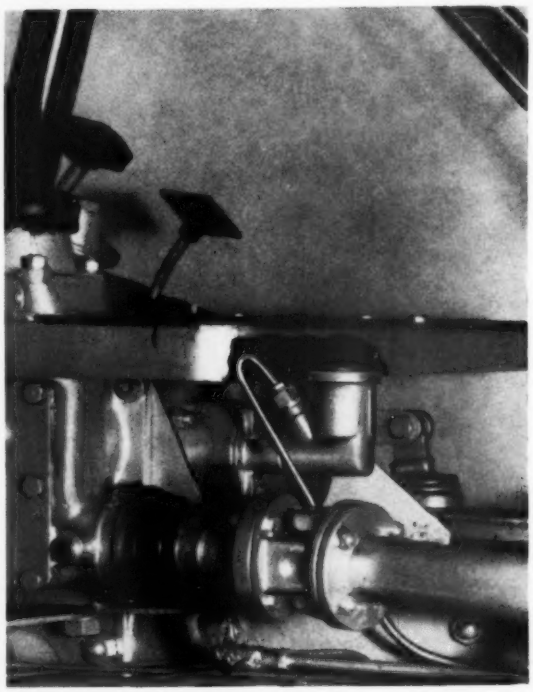

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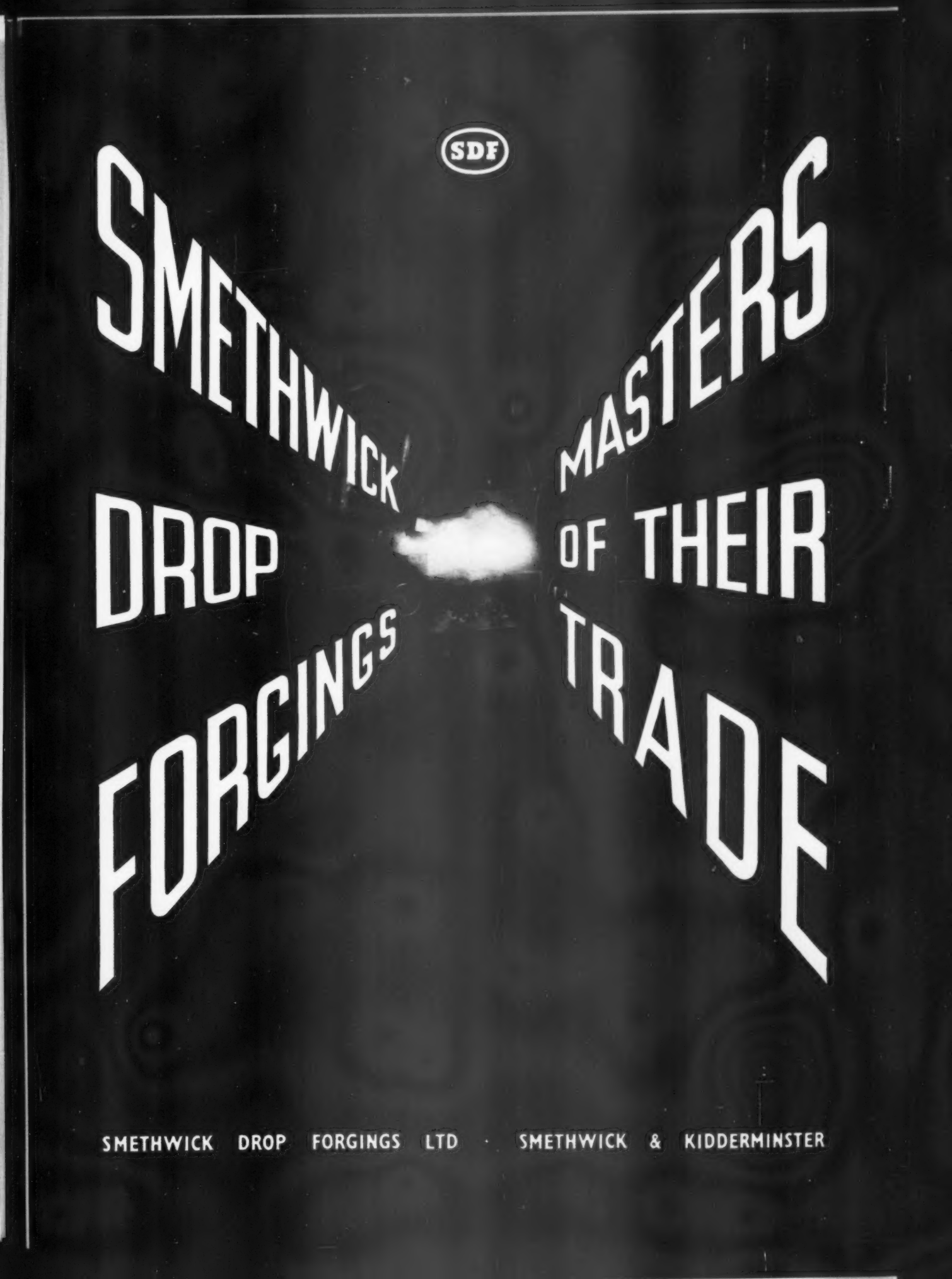
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AP 38

AUTOMOBILE ENGINEER

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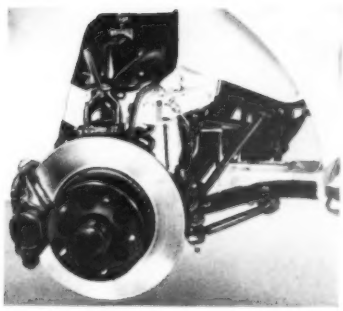

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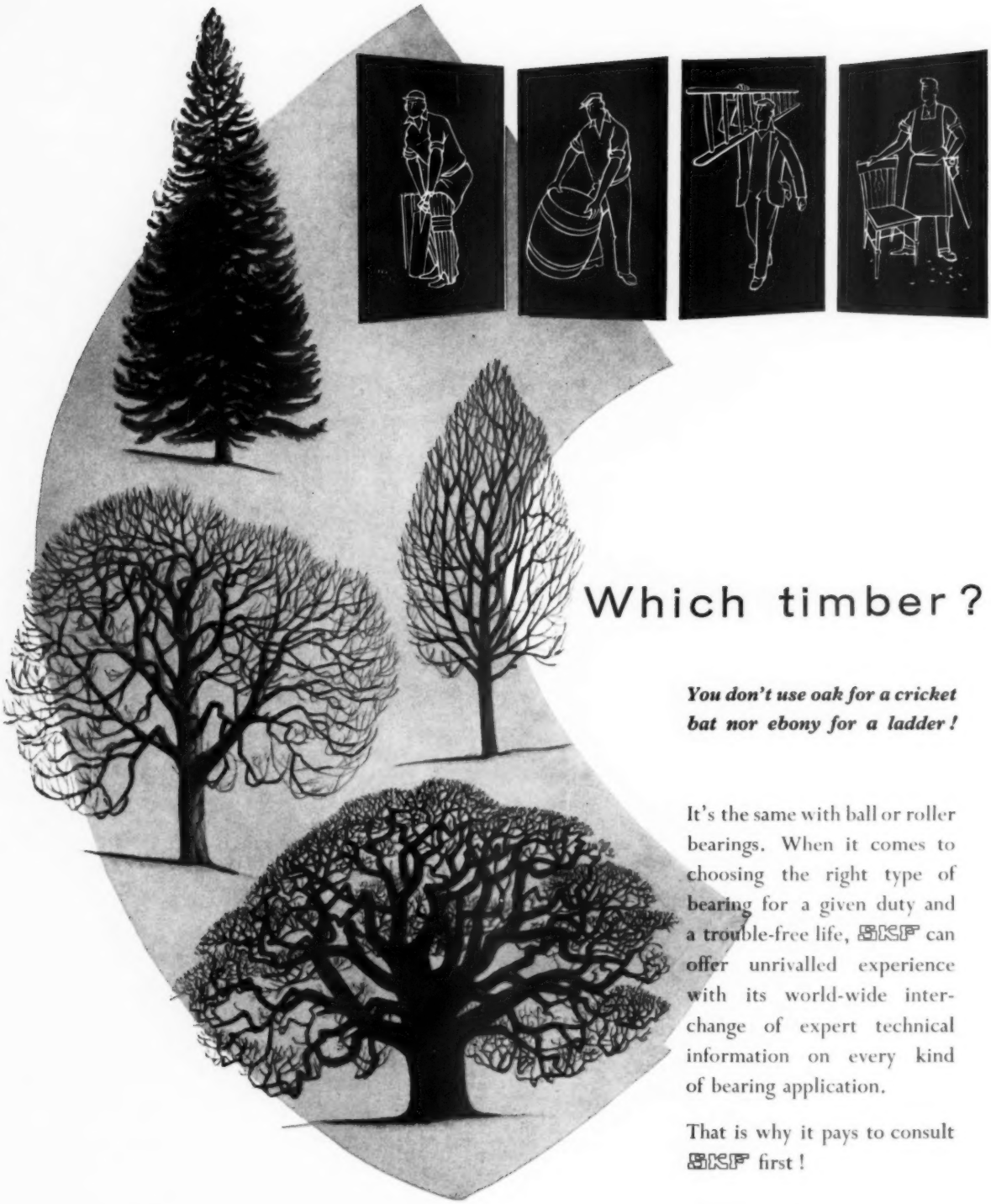

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DESIGN MATERIALS **AUTOMOBILE ENGINEER** PRODUCTION METHODS WORKS EQUIPMENT

Aluminium Engines

IT is well known that, in Great Britain, the United States and the motor manufacturing countries on the Continent, intensive investigations are in progress in respect of the employment of cylinder block and crankcase units made of aluminium alloy. By far the greatest amount of publicity on this subject has, of course, been issued by American firms. The principal advantages of aluminium for this application are its low weight, high thermal conductivity and the possibility of manufacture by die-casting, and thus eliminating practically all the subsequent machining operations.

It is probable that the weight of an automobile engine with an aluminium cylinder head and cylinder block and crankcase unit would be approximately 30 per cent less than that of a similar assembly of conventional cast iron components. This weight reduction has secondary beneficial effects: for example, the overall weight of the complete structure could be less, the brakes of the vehicle could be smaller and lighter, and acceleration and fuel consumption would be improved. Also, because of the relatively high thermal conductivity of aluminium, problems with regard to the prevention of knock and pre-ignition can be more easily solved.

It is well known that the principal disadvantages of aluminium relative to cast iron are its high cost per pound, low value of Young's modulus, large coefficient of thermal expansion and its tendency to corrode in certain circumstances. Obviously, there are also problems with regard to wear resistance. Nevertheless, the advantages of the material are so great that it is well worth while to devote considerable resources to overcoming the disadvantages.

The high cost of the metal can be offset by other economies. Provided gravity or pressure die-casting can be employed, most of the subsequent machining operations that are necessary on cast iron blocks can be avoided with aluminium. The wall thickness of die-cast aluminium components can be less than that of iron ones, since the minimum permissible dimensions are not likely to be governed by practical requirements in respect of casting technique. In this connection, however, the relatively low stiffness of aluminium must also be considered. Further economies could be made by casting the clutch and gear cases integrally with the crankcase. This would also help to overcome problems with regard to stiffness.

Difficulties due to thermal expansion can be obviated, or at least reduced, in two ways. First, aluminium alloys with a high silicon content can be used, since they have a lower rate of expansion than most. The second measure that might be adopted is the incorporation of steel inserts in the critical sections, such as those that house the main journal bearings. This composite type of construction has, of course, already been used for pistons.

Corrosion is difficult to combat, and cavitation is also likely to be encountered. Probably the most promising line of approach is the development of suitable corrosion inhibitors. It should not be difficult to spray anti-corrosion coatings on to the internal surfaces of die-castings. In particular, it is necessary to pay attention to the prevention of corrosion between joint faces and in other small crevices. Fortunately, the alloys with a high silicon content are not so susceptible to corrosion as are some of the others. The effects of alkaline salts, which are present in the water in certain localities, and of anti-freeze mixtures, also have to be studied carefully in relation to the effectiveness of various inhibitors.

The wear-resistant aluminium silicon alloys have been reported to be even less prone to scoring than is cast iron. However, the tests on which these reports are based seem to have been carried out in a laboratory, and so the question arises as to whether the results are applicable to practical conditions. Unfortunately, most of these alloys are relatively difficult to machine and to cast, but machining operations can be largely obviated, and improved techniques may make casting practicable. Apparently, there are no lubrication problems, possibly to a large extent because of the high thermal conductivity of aluminium. However, a field that ought to be explored by the oil industry is the formulation of lubricants and additives that are better suited to engines of aluminium construction than are those currently in use.

Obviously, the designs of engines initially introduced will be greatly influenced by current practice in respect of more conventional units, and also by existing manufacturing facilities. Subsequent experience with this new type of engine, and further advances in the metallurgical and production fields, will certainly result in development of designs that in certain details are markedly different from those in use at present. It is confidently expected that if aluminium is adopted for quantity produced engines for cars and commercial vehicles, the main components eventually will be produced by die-casting. At least two manufacturers in this country have equipment large enough for the production of die-castings for engines up to about 1 litre capacity, and other machines are being made elsewhere for the production of castings for engines with cylinders of up to about 2½ litres swept volume.

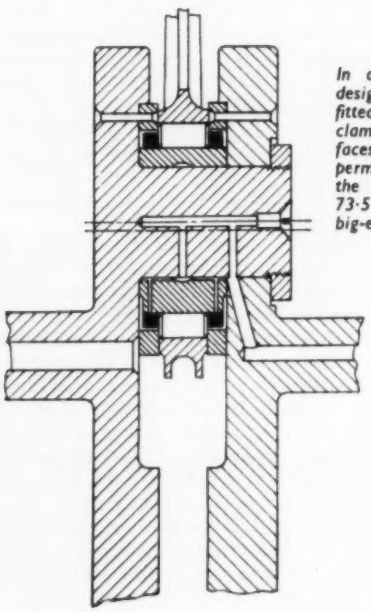
NORTON ENGINE DEVELOPMENTS

Some Interesting Design Activities on Racing Motor Cycle Power Units

ALTHOUGH the racing history of Norton Motors Ltd., of Birmingham, goes back to the early years of the century, it is during the past 30 years that the company's long list of international successes has been amassed. Throughout that period, technical progress on the racing engines has been maintained at a remarkably high level, and the power outputs of the world-famous 350 and 500 cm³ units have risen steadily. An indication of Norton Motors' knowledge in this field is their claim to have been the first to develop a half-litre single-cylinder engine, with racing reliability, to exceed 100 b.h.p/litre swept volume, on a normal hydrocarbon fuel, that is, not an alcohol-base fuel.

Several years have now elapsed since the company withdrew its official support of grand prix motor-cycle racing. However, it has continued its relatively small-scale manufacture of specialized racing machines for normal sale. During the past year, the development programme for the engines of these Manx machines, as they are called, has been particularly intensive, and several design features of considerable technical interest have emerged.

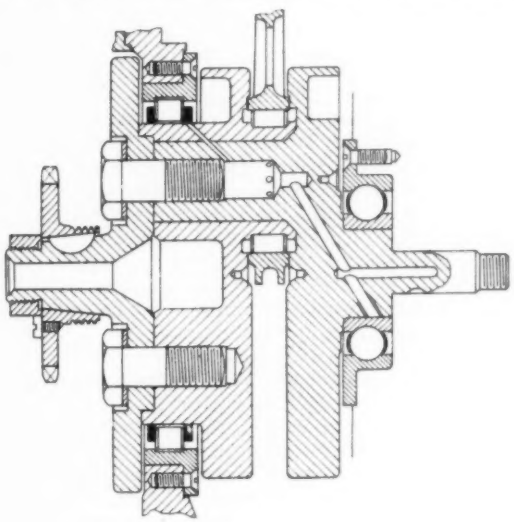
The aim throughout this recent programme has been at increasing the output, by enabling the engine to run at higher r.p.m. As would be expected, the only way of obtaining any substantial increase in rotational speed was to shorten the stroke, and two of the lines of development have been in that direction. In the 500 cm³ class, success had been gained previously with engines having a stroke of 78 mm, in conjunction with a 90 mm bore, but the still lower stroke:bore ratios envisaged were not possible with the existing crankshaft layout. It was desired to retain a roller type big-end bearing, because it combines low friction with a favourable ratio of width to load-carrying capacity, thus making for a stiff shaft. A one-piece crankshaft would have necessitated a split big-end bearing, which was not regarded as entirely satisfactory for the arduous duty. The built-up crankshaft, therefore, had to be retained also, but with the existing construction there was a minimum practicable stroke, because of the need for clearance between one main bearing, or its housing, and the nut securing the crankpin in the flywheel.



In one Norton crankshaft design, an eccentric sleeve, fitted on the crankpin and clamped between the inner faces of the flywheels, has permitted a reduction in the stroke from 78 mm to 73·5 mm. A roller type of big-end bearing is employed

Two ingenious methods were evolved of overcoming the difficulty with regard to clearance. So far, only the first of these has actually been used to obtain a reduced stroke, but it was only intended as a temporary expedient to enable a shorter-stroke engine to be built and run with the minimum delay. It was, in fact, embodied in an engine that was raced with promising results in 1958. This layout is remarkably simple and has as its basis the crankshaft assembly of the 90 mm bore engine, on which the crankpin is integral with one of the flywheels. The crankpin was reduced in diameter and an eccentric sleeve was pushed over it. Then the other flywheel was pressed on to the shaft and its securing nut tightened to clamp the sleeve in position. Drilled holes, and grooves in the sleeve, carry oil from the existing lubrication passages in the shaft to the cage and rollers. The sleeve was case-hardened and ground to form the inner track for the rollers. With the degree of eccentricity used, the minimum stroke was reduced from 78 to 73·5 mm; with a 93 mm bore and this stroke, a swept volume of 499 cm³ was obtained. The big-end dimensions were the same as those of the standard engine, but the dynamic mass was reduced by fitting a shorter connecting rod, and the flywheels were rebalanced accordingly.

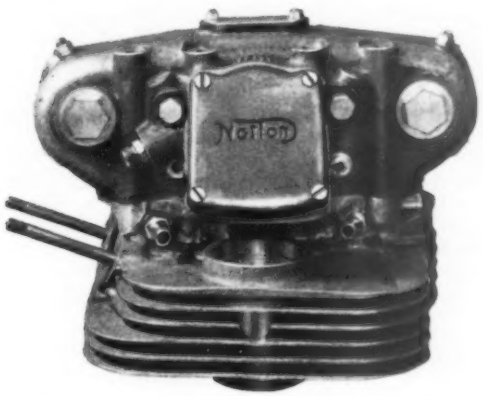
A still shorter stroke can be obtained with this entirely different crankshaft layout. Its unusual features include a very large roller bearing on one side of the shaft, together with a bolted-on external flywheel



Concurrently with the development of the eccentric-sleeve crankshaft, a completely different and more robust method of shaft construction was evolved, with a longer-term policy in view. To enable a direct comparison to be carried out with the crankshaft of the production Manx engine, the new assembly was designed with the standard

stroke of 85·62 mm. However, the design principle permits a considerable reduction of the stroke, below that practicable with the eccentric-sleeve construction.

The scheme is noteworthy for the use of a three-piece crankshaft and a very large roller main bearing to carry the drive end of that shaft. This bearing has a diameter appreciably greater than the stroke, so that its overall radius is greater than that of the crank throw plus the radius of the pin. As can be seen from the accompanying illustration of the layout, the crankpin is hollow and is forged integrally with the internal flywheel on the side from which the drive is taken. On the outboard side of this flywheel is the short, 5½ in diameter mainshaft, the surface of which is hardened to form the inner track for the rollers of the main bearing.

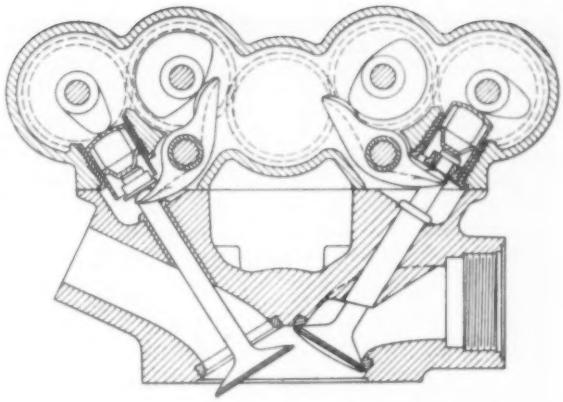


Apart from the absence of valve springs, the housing for the cams of the desmodromic valve actuating mechanism has an orthodox appearance

A second forging in the assembly comprises the internal flywheel nearest the timing drive, its mainshaft and a long, internally threaded spigot, which is a press fit in the bore of the crankpin. There is a conical seating face at the root of the spigot and in the end of the crankpin bore. The third component of the crankshaft assembly, also a forging, is an external flywheel integral with the shaft carrying the chain-sprocket of the primary drive to the gearbox. By the use of an external flywheel to supplement the internal pair, the dimensions of these, and, therefore, of the crankcase, can be kept to a minimum. On the inner face of the external flywheel is a shallow spigot that registers in a counterbore in the mainshaft. Through the spigot pass the four bolts holding together the three components of the crankshaft. One bolt screws into the long spigot within the crankpin, while the other three secure the external flywheel to the internal flywheel on the side from which the drive is taken.

To obviate any possibility of the outer race of the large main bearing turning in its boss in the crankcase wall, it is flanged at its inner end; countersunk screws secure the flange to the boss. A similar precautionary measure is employed for the outer race of the normal-size ball bearing that carries the mainshaft nearest to the timing drive. On the other side, a cover plate, which closes the housing for the external flywheel, is bolted to the exterior of that half of the crankcase; there is a running clearance between it and the boss of the taper-mounted chain-sprocket on the drive shaft. Outboard of the large bearing, a spring-loaded, lipped oil seal prevents leakage from the crankcase into the cover; for simplicity, this seal is not shown in the illustration.

When this design was first envisaged, it was realized that the high peripheral speed of so large a bearing would demand unusually high standards of accuracy, both in the manufacture of the bearing itself and in its installation.



A train of five gears is employed to operate the separate opening and closing cams. The opening tappets are of inverted bucket type, and the closing cams actuate levers that bear on collars on the valve stems

Also, there was the possibility that, in spite of the relatively low loading per roller, this high speed might restrict engine r.p.m. to a lower figure than would otherwise be possible. The problem was referred to the Hoffmann Manufacturing Co. Ltd., of Chelmsford, who proposed a bearing based on one that had already been used successfully for gas turbine rotors and, therefore, had proved its reliability at speeds higher than were contemplated for this application.

The lubrication of this bearing was clearly a matter of vital importance, and much care was taken by Norton Motors to arrange the best possible oil feed to it. From the pump, the oil is fed to a hole drilled in the mainshaft on the side adjacent to the timing drive, and thence to a chamber in the crankpin spigot, from which other drilled ducts take it to the rollers of the big-end bearing and the cage of the main bearing. Any substantial reduction of the stroke that may be made in the future will, of course, enable a smaller bearing to be used, with a consequent lowering of the peripheral speed at any given crankshaft r.p.m.

Whereas the arrangement, as illustrated, permits the use of an appreciably shorter stroke than has hitherto been employed, variations are obviously possible that would enable the stroke to be shortened further if desired. The hexagon-head bolts that secure the crankshaft assembly could presumably be replaced by the socket type, to permit them to be arranged on a smaller pitch circle, or the threaded hole could be eccentric in the crankpin spigot. If the mainshaft were hollow and of sufficient diameter, the crankpin bolt could be inserted through it, and in this way the stroke could be brought down to any desired figure.

The third Norton design of interest is a desmodromic system of valve actuation. Although technically sound, it has not yet been developed sufficiently to show marked superiority over the orthodox valve gear, with two overhead camshafts, which has years of racing experience behind it. Externally, the cam box of the desmodromic gear differs from that used in the Manx engine mainly in respect of the absence of the exposed hairpin valve springs that have for so long been a characteristic of the Norton racing engines. As before, the box contains a train of five gears, the middle one of which is driven by a bevel gear on the vertical shaft. Again as hitherto, the opening cams are mounted on the outboard gears of the train, and actuate the valves through the medium of inverted bucket type tappets having flats on their sides, to prevent rotation, and arcuate follower surfaces.

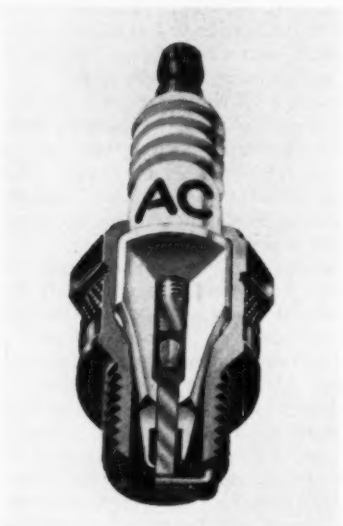
On the conventional valve gear, the intermediate gears are merely idlers, but on the new assembly they carry closing cams of the usual complementary form. These cams

actuate rockers, the spindles of which lie immediately below the cam spindles. The other arms of the rockers are forked and bear on collars on the valve stems; these collars are secured by split collets in the manner used for retaining valve spring collars.

Initially, the same valve opening diagram was employed on the desmodromic mechanism as on the standard Manx engine. The object at that time was to compare friction losses in the two systems, rather than to raise the output. Later, the rates of acceleration and deceleration were increased considerably, with the result that the vertical drive shaft proved inadequate for the higher loadings, and failure of the Oldham couplings occurred. A new drive assembly was, therefore, designed; in it the upper and lower bevel gear shafts are each supported in a pair of needle-roller bearings, and have splined ends coupled by an internally splined sleeve. The splines are a push fit to allow for differential thermal expansion. Since the new drive proved capable of withstanding the exceptional loads imposed by the desmodromic valve gear, it has been adopted as standard on the 1959 production racing engines, on which the normal valve gear continues to be employed.

New AC Sparking Plugs

A COMPLETELY new range of AC sparking plugs, designated the Hot Tip series, has recently been introduced by the AC-Delco Division of General Motors Ltd. The main difference between these new plugs and their predecessors is in the insulator and electrode assembly, which has been completely redesigned. As before, sintered aluminium oxide is used for the insulator, but improvements have been made to the manufacturing processes, with the result that the properties of the material have been enhanced. The name Hot Tip has been adopted because of the special design of the insulator nose, which is recessed round the electrode and of very thin section. As a result it responds rapidly to changes of temperature and thus tends to follow closely the temperature cycle within the cylinder. The consequently high temperatures reached during combustion ensure that carbon and oil deposits are burnt off as they form. On the other hand the insulator cools sufficiently between combustion cycles to avoid pre-ignition. There is ample space between the insulator nose and the bore of the threaded portion of the plug to permit scouring by the hot gases.



This sectioned view of one of the AC Hot Tip sparking plugs shows the recessed nose of the insulator and the novel copper-ceramic type of electrode seal

The second important feature of these plugs is the use of a fused copper-ceramic electrode seal, known by the trade name of Conductaseal. This material provides the necessary high electrical and thermal conductivities between the upper and lower portions of the electrode and forms an effective bond between these metal portions and the insulator material.

Formed on the upper portion of the insulator exterior are four circumferential buttress ribs, which are precision ground to a carefully chosen profile. The buttress shape not only reduces the likelihood of flash-over, by lengthening the electrical path, but it also has an umbrella effect that reduces the accumulation of dirt and moisture, which are conducive to electrical tracking.

By virtue of the Hot Tip principle, the plugs have a greater heat latitude than their predecessors. The range marketed covers 10, 14 and 18 mm threads, in a variety of reaches and heat values suitable for all commonly used types of engines. An application and conversion list is available from the makers at their Dunstable, Beds., factory.

Jointing Compound

NORMAL jointing compounds have proved unsuitable for many gas turbine requirements, because of the special lubricants used and the high operating temperatures and narrow joint faces involved. In consequence, a special compound with a polyester-urethane basis has been developed by Rolls-Royce Ltd., and is now manufactured under the trade name Hylomar SQ32, by Marston Lubricants Ltd., Rock Ferry Oil Works, Birkenhead. Rigid standards for the specification and manufacture have been laid down by Rolls-Royce to ensure consistency of the product.

The properties of Hylomar compound give it a very wide field of application. It has a temperature range of -50 to $+300$ deg C, is non-inflammable, harmless to all metals, and is resistant to water, glycol, gasoline and kerosene fuels, petroleum and synthetic lubricants, and to products of combustion. Its physical properties are such that, during assembly of the components, plastic flow occurs, leaving between the faces the minimum quantity necessary to fill up surface irregularities. The tackiness of the compound prevents any squeezing-out of this minimum quantity and so avoids the need of subsequent retightening of the securing nuts or bolts in service.

Application of the compound is made to both faces of the joint. To allow the development of the necessary tackiness and adhesion to the metal, an air drying period of at least 10 minutes must then be allowed before the faces are united. A valuable characteristic is that the delay between coating and bolting-up can be extended indefinitely without detriment to the sealing properties. The compound can be removed from dismantled joints by rubbing with a cloth soaked in trichlorethylene.

To suit various conditions of usage, four grades of Hylomar are manufactured: stiff, heavy, medium and light. The stiff grade has to be applied with a knife, but the others can be brushed on to the surfaces. Packaging of the three lighter grades is in 4 oz collapsible tubes, while the other is supplied in tins. A brochure giving useful additional information is available on request from Marston Lubricants Ltd.

Left: The use of silicone release agents has been found beneficial in preventing adhesion between tyres, or other rubber components, and their moulds. Right: Silicone greases are widely employed to protect ignition systems from the effects of moisture or even of large quantities of water

Automobile Applications of Silicones

A Review of Current Usage of Silicone Fluids, Resins, Greases and Rubbers

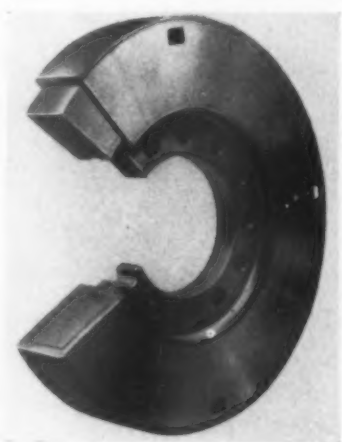
DURING recent years, the use of silicone compounds has increased enormously in a wide variety of industries, and most engineers are familiar with some of the properties of these compounds. Probably the water repellency is the most spectacular, but the insulating ability and the resistance to extremes of temperature are also generally known. These and the other unusual properties of silicones make them suitable for a number of automobile applications, both on vehicles and in their manufacture.

Silicones are semi-organic polymeric materials based on chains of alternate silicon and oxygen atoms. These linkages are also the basis of the chemical structure of glass and mica but those materials are intractable and brittle because the linkages form a random network. In the silicones, organic groups such as methyl, phenyl or vinyl are also present and are attached to the silicon atoms. Variations in the properties of the material are obtained by varying the organic groups and controlling the linking together of the silicon-oxygen chains.

The most common silicones are the fluids, so it is logical to discuss them first. They are used to treat textiles and leather, to render them water repellent without impairment of the breathing properties. Thus there are silicone applications in the upholstery and trimming of cars. However, the use in this field is relatively restricted because silicone treatment does not greatly improve the material's basic resistance to staining by oil or grease, whereas the various plastics fabrics can readily be cleaned of such stains. It is probable, though, that leather will continue to find favour for at least the upholstery of the higher-grade cars, and its treatment with a silicone could be particularly beneficial in the case of an open or drop-head body.

Silicone fluids have a very low viscosity-temperature coefficient. In addition they have a high resistance to shear and are generally inert, a combination which renders them very attractive for damping purposes in mechanisms. A common use in this connection is in the pivot bearings of pointers of magnetic speedometers and tachometers, to prevent fluttering and permit more accurate reading. In the Holset torsional vibration damper, fitted to both diesel and petrol engines, a silicone fluid of very high viscosity is used as the damping medium. A familiar application of the lower-viscosity fluids is in body polishes; the addition of silicones imparts a degree of water repellency but probably their main benefit is the ease of application.

The power absorption and noise of engine cooling fans have been causing some concern. To deal with these



A silicone fluid of very high viscosity comprises the damping medium in the Holset torsional vibration damper fitted to a variety of engines

problems, two types of fan coupling embodying silicone fluids have been introduced in the U.S.A. The earlier of these, a product of the Schwitzer Corporation, is used on Mercury cars and proportions the fan speed to the cooling needs of the engine. Thus it minimizes power wastage when the coolant temperature is low, and also it prevents excessive fan speed, and therefore noise, when the engine is running fast under light load. In the latter conditions, the power saving can be 80 per cent or more.

Within the coupling are adjacent driving and driven faces separated by the silicone fluid; the distance between the faces is controlled by a thermostat. When the coolant temperature is below 160 deg F, the distance is such that the viscosity of the fluid can only exert limited drag between the faces. In consequence, the fan speed is held down to 1,500 r.p.m. As the temperature rises, the thermostat brings the faces closer and the increased drag causes the fan speed to increase. However, the gap is still sufficient at nearly boiling temperature to prevent the fan speed from exceeding 2,600 r.p.m. Owing to the thermal stability of the fluid, the characteristics of the coupling are virtually unaffected by variations in ambient temperatures.

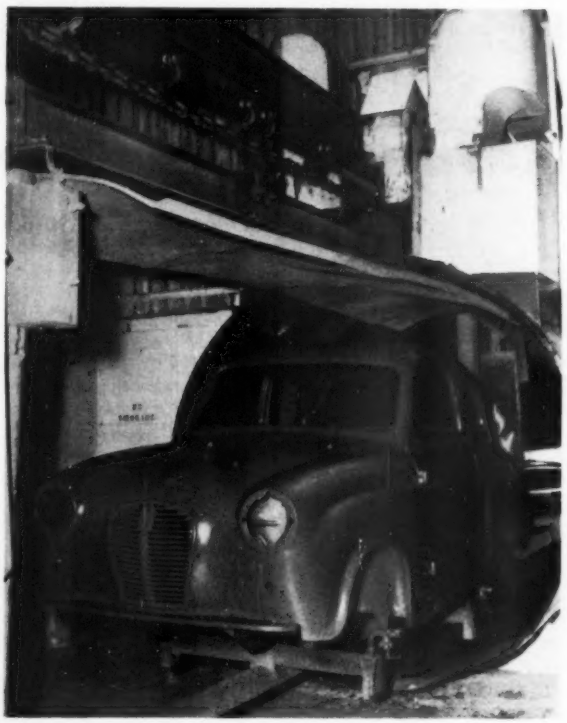
The second coupling was designed by the Eaton Manufacturing Co., and its manufacture in this country is being considered. It, also, works on the principle of torque transmission through viscous shear of the silicone fluid, but it is simpler in that there is no thermostatic control. In effect, the device is a torque-sensitive clutch whereas the Schwitzer coupling is sensitive to temperature.

Coupled to the belt pulley is a circular plate member having a number of concentric, shallow ribs on one face, to increase the surface area. The plate is enclosed within a light alloy casing to which the fan is attached. On one inner face of the casing are shallow grooves of similar form to the ribs just mentioned. The small clearance between the driving and driven members is filled with the silicone fluid.

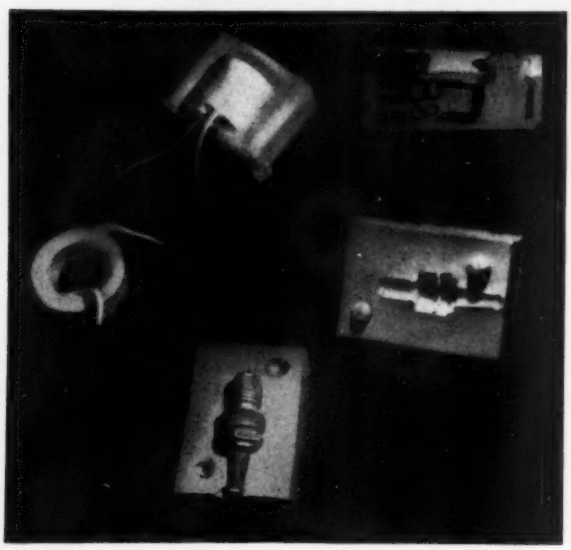
To give adequate engine cooling at low speeds, a larger fan or higher drive ratio can be employed than is normally the case, without the disadvantage of excessive power consumption and noise at higher speeds. As the fan accelerates with a rise in engine speed, its resistance to rotation increases, so that slip begins to occur as a result of shear of the fluid. The slip increases progressively as the input speed to the coupling rises, and beyond about 50 per cent of maximum engine r.p.m. no further increase in fan speed occurs.

Some power is obviously dissipated as heat, owing to the work done in shearing the fluid, but even so the total absorption at high engine speeds is only in the region of a fifth of that of a normal direct-drive fan. Since the fan speed is usually limited to about 2,000 r.p.m., the noise is restricted to a much more acceptable level, possibly 20 decibels lower than with an orthodox drive at maximum input speed. As in the previous case, consistency of performance with variations in ambient conditions is ensured by the flatness of the viscosity curve of the fluid. The Eaton coupling is compact and simple to manufacture, and it will be interesting to see what response is made to it by British engine manufacturers.

An important industrial and naval application of silicone resins is for electrical insulation but, so far, little has been done regarding their introduction on automobile electrical equipment. However, the high heat resistance of these resins can be taken advantage of in paint manufacture: continuous operating temperatures as high as 600 deg C can be withstood by paints made with silicone resins. This feature should render them suitable for exhaust systems, on which the majority of other finishes do not prove adequately durable.



Right: Silicone grease is used for lubricating the conveyor chains of this infra-red, paint stoving plant. Below: Cold-curing silicone rubber forms a useful means of protecting delicate components or equipment



Because silicones are resistant to low as well as high temperatures, greases embodying them can be used for a large variety of duties, some of which would be beyond the scope of petroleum-base greases. The operational temperature range of such lubricants is from -70 to $+200$ deg C. One well-known car manufacturer uses a silicone grease for the conveyor chains of several high-temperature paint stoving ovens in the body shop. The use of such a grease has considerably extended the life of the chains.

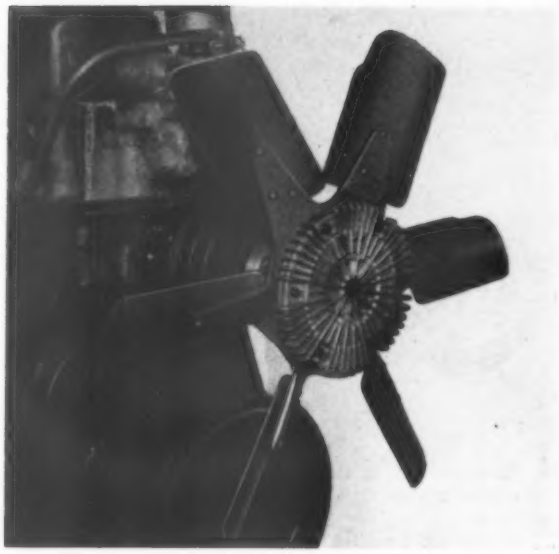
Another form of silicone grease is not intended primarily for use as a lubricant, but is valuable as an insulating

compound for electrical systems. Because of its good dielectric properties, water repellency, freedom from oxidation and ability to withstand high temperatures without melting, it is widely used as an insulant on aircraft and on fighting vehicles. Smeared thinly on ignition leads, at their entry points on the distributor and coil, and on sparking plug insulators, the substance has been found most effective in keeping out moisture.

Though expensive in terms of cost per pound, this grease is economical in use and could undoubtedly be of benefit to those many car owners whose vehicles have perforce to stand in the open in all weathers. Much of the starting difficulty frequently encountered with such vehicles is, of course, caused by moisture condensation on the ignition equipment. The grease can also be used for the coating of rubber weather-sealing strip on vehicles, to prolong its life and to prevent freezing or sticking; also as a non-freezing door lock lubricant and as a protective against the corrosion of battery terminals.

It has already been indicated that silicones have anti-adhesion properties, and these have been found beneficial in a number of industries associated with the manufacture of motor vehicles. Silicone release agents are used to prevent tyres and other rubber components sticking to their moulds, and they serve a similar purpose in plastics factories, and in some foundry work, notably shell moulding. Not only are silicone release agents unaffected by the temperatures encountered in the applications mentioned but they assist in providing a good surface finish on the article being produced. So far, little use has been made of the silicone rubbers in the automobile sphere, partly because the superior heat resisting and electrical properties of silicone rubbers are not usually a necessity but there are many applications where they would be a decided advantage. Electric cables so insulated should last as long as the car, even in under-bonnet conditions, and a variety of boots, gaiters and seals could with advantage be made in these materials.

In aircraft work, where primary cost is less important than suitability for the function, the silicone rubbers are widely employed for a great number of applications. Their elastic range of -90 to $+250$ deg C renders them particularly valuable for high-altitude and very high speed conditions. Small, special purpose flexible heaters are produced by embedding a mat of resistance wire and woven glass in silicone rubber. Again, a recently developed transparent silicone rubber makes an excellent interlayer for aircraft windscreens to prevent splintering. Organic plastics materials usually used as interlayers will not withstand the temperatures encountered in very high speed aircraft owing to frictional heat. As a cable insulant, silicone rubber has been found to be particularly satisfactory for shipboard communication and power cables owing to its heat resistance, excellent electrical properties and its ability to function after fire. This is one of the main reasons why the Admiralty has decided that all electrical wiring in H.M. ships will in future be carried out in silicone rubber insulated cable.



An interesting American development is this Eaton fan drive coupling that utilizes silicone fluid to provide slip as the engine speed rises

Cold-curing silicone rubber is a recent development of interest. This material possesses many of the properties of the hot-curing silicone rubber and, in much the same way as with polyester resins, is cured with a catalyst which is added just before use. The setting time depends upon the quantity of catalyst added and can be varied from a matter of minutes to hours. Rubbers of this type are particularly suitable for capsules for the enclosure of delicate electronic equipment, which can be carried out on the spot, for moulding purposes or for making capsules to hold such components as sparking plugs.

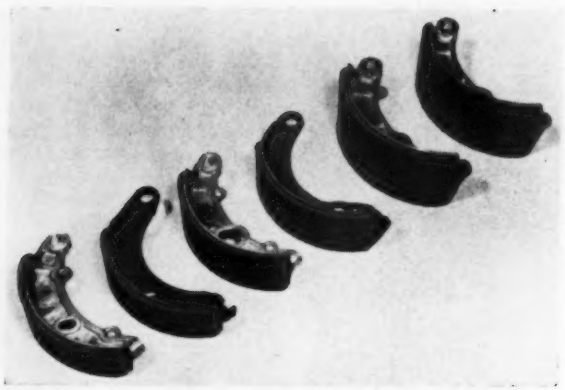
Finally, mention must be made of the silicone anti-foam agents, available as a compound, a fluid or an emulsion, depending on the duty. The killing of foam can often result in the speeding-up of an industrial process, and the silicone anti-foam agents are claimed to be superior to others in their versatility and economy in use. To take examples connected with motor vehicles, foam is generated in the manufacture of rubber, paints, resins and lubricating oils, and silicone anti-foam agents are used successfully in each instance.

Most of the information from which this article was compiled was supplied by Midland Silicones, Ltd., 68 Knightsbridge, S.W.1, and their associated U.S.A. company, Dow Corning Corporation, Midland, Michigan.

Pressure Transducer

AN AGREEMENT has recently been concluded whereby the Solartron Electronic Group Ltd., Thames Ditton, Surrey, has acquired the rights to manufacture and market a new pressure transducer of vibrating cylinder type, designed by the Svenska Flygmotor Aktiebolaget. Production has already begun and the transducer has Ministry of Supply approval for incorporation in missiles and other equipment during test.

The pressure sensitive element of the transducer is a mechanical vibrating system, the natural frequency of which depends on the pressure applied. This vibrating element controls the frequency of an electronic feed-back oscillator thereby providing an output voltage of constant amplitude but of a frequency that varies with the applied pressure. It is stated that the variable frequency of the output signal makes the transducer especially suitable for magnetic tape recording, and also for analogue-to-digital conversion by pulse counting. An advantage is that the equipment utilizing the signal from the transducer can, without loss of accuracy, be located a considerable distance from the point of measurement. Satisfactory results have been obtained even with a distance of several miles between the two components.



BONDING OF BRAKE LINERS

Description of Methods Used in Manufacture and in Servicing

R. A. JOHNSON*

THE advantages of using synthetic resin adhesives for the attachment of brake liners to brake shoes are now well known and appreciated. Briefly they are:

1. Greater efficiency, owing to the absence of rivet holes and, consequently, larger rubbing area of liner.
2. Freedom from scoring of brake drums.
3. Better heat dissipation, since the liner is in intimate contact with the shoe.
4. Increased life of liner because it may be worn almost to the shoe and effective thickness of material is not restricted by the rivet heads.

Large scale production experience during the last few years has now proved another advantage: it is that the cost of bonding liners to shoes can be as much as 20 per cent less than riveting. Bonding has been widely used in the American automobile industry for many years, but only in the last four or five years have the European manufacturers made large scale use of adhesives for application to road vehicles in relatively large quantity production.

One of the largest brake bonding plants in any European automobile factory is to be found in the works of Daimler-Benz A.G., of Stuttgart. Their interest in bonded brakes originated during the days of their success in the field of motor racing, when they used Redux 64 adhesive to help to obtain high braking efficiency. Not until the adhesive had proved itself in the exacting conditions of the race track was bonding considered seriously for production cars. Since then several million brake shoes have been bonded, with complete success from both the production and service points of view.

This illustration shows bonded brake shoes being removed from the curing oven at the factory of Daimler-Benz A.G. at Stuttgart

Large-scale production

One of the most important considerations when using any form of adhesive is that the surfaces of the parts being bonded should be suitably and efficiently prepared. In the Daimler-Benz factory, the preparation is as follows. First, the bonding surface of the brake shoe is roughened by shot blasting. It has been found the type of shot used is important, the best results being obtained with a very sharp shot. Next, the liner surface is roughened by sanding; in this operation it is necessary to ensure that any identification marks are completely removed from the surface. After the roughening processes, both liner and shoe are degreased in a solvent vapour-bath, the solvent most commonly used being trichlorethylene.

Next in the process is the application of the adhesive, in this case Redux 64. This is a liquid, modified phenolic adhesive containing solvents. At the works of Daimler-Benz, it is applied to both liner and shoe, a measure which, although not absolutely necessary, indicates the thoroughness with which this company treats the process of bonding. The method of application is by brush or spray gun, after which the solvents are dried off. Evaporation of the solvents must take place before the liner and shoe are brought together; at Stuttgart this is accelerated by placing the coated parts in a suitably ventilated oven at about 80 deg C.

When the solvents have evaporated, the adhesive becomes dry and, in this condition, the parts may be kept for several weeks before completing the curing process. However, in a production sequence they are, of course, kept for only a few hours before being clamped together and heated under pressure, to effect the curing of the adhesive. In the Daimler-Benz factory, the shoes and liners are assembled, in pairs, by

*CIBA (A.R.L.) Limited.

placing them inside steel bands and applying pressure by expanding the shoes against the steel band, by means of a screw jack as shown in an accompanying illustration. The assembled bands are then fed into an oven on a conveyor belt and the adhesive cured. This may be done at any temperature between 145 deg C and 200 deg C, the length of time required being in inverse relation to the temperature. At 150 deg C the time required at that temperature is 30 minutes, whilst at 180 deg C it is only 4 minutes. Extra time must however be allowed for the brake and band assembly to reach the curing temperature.

After curing is completed, the assembly is removed from the oven and is usually allowed to cool before pressure is released. Any spewed adhesive is then removed, after which the strength of the bond may be tested. At Stuttgart, this is done by carrying out a shear test to destruction on one corner of every shoe bonded.

There are other methods of applying the necessary heat and pressure: in one system that is used extensively, pre-heated anvils are employed in conjunction with pressure clamps, which are usually pneumatically operated. A typical system of this type has electrically heated anvils, the diameters of which are arranged to suit the inside diameter of the brake shoe, each with a slot into which the web of the shoe can fit. The electrical heating elements are usually embedded in the anvil at a constant distance from the working face and are heated continuously to maintain a fairly constant face temperature. Pressure is applied by means of a steel band or pressure plate, which is brought down on to the brake lining after the shoe and liner have been assembled and placed on the anvil. The most common method of applying the pressure is by air operated clamps controlled by a pedal, to

Screw jacks are used to expand the brake shoes and liners against the steel bands before inserting them in the curing oven at Stuttgart

Redux 64 being brushed on to the shot-blasted and degreased metal surface of the shoes; the solvent is dried off before the liner is fitted

leave the operator's hands free for loading and unloading.

In one such system which is used to cure Redux 64, the anvil is maintained at a temperature of approximately 280 deg C and the shoe and liner, which are both at room temperature when inserted, are kept under pressure for a period varying from 2 to 3 minutes according to the thickness of the shoe. After this time, the hot assembly is removed and placed aside to cool. When cooled, the Redux 64 is completely cured and no further oven curing is required. With this type of curing, one operator can look after several anvils, and labour time is less than one minute per assembly.

The re-lining of bonded brakes

It is expected that every liner will be replaced at least three times during the life of the vehicle; in the case of heavy transport vehicles the number of replacements will be considerably larger. The importance of relining of brakes is therefore equal to that of the manufacture of the original assembly. An example of the way in which a completely new industry has arisen for this purpose is shown by the activities of Société Européenne de Bonding, whose head-quarters are in Paris.

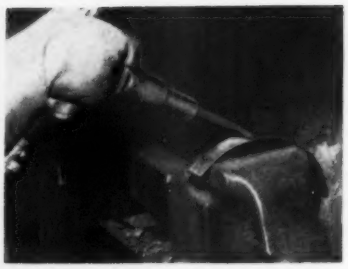
This company was formed some years ago to promote the re-lining of brakes by adhesives. After testing all the available adhesives, they finally built up their methods around the use of Redux. The Société has now been responsible for the setting up of about one hundred and fifty bonding stations throughout France. It is now extending its operations, through the medium of associated companies, to include the Netherlands and Germany. Whilst its methods of bonding are basically similar to those already described, there is the additional and important task of removing the worn liner before it is possible to begin the sequence of operations for attaching the replacement liner.

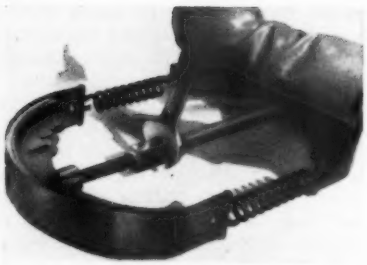
In the beginning, various ways of removing worn liners were tried. These were:

(a) Peeling of the liner by means of what may best be described as a lathe turning operation in very slow motion followed by removal of the last very thin layer by linishing.
(b) Burning off the worn liner by putting the shoe into a furnace.
(c) Removing the bulk of the worn liner by means of an air operated chisel; this method is the one that was finally adopted for regular replacement production.

The shoe is held in a vice and the liner removed by a chisel that is at least as wide as the liner. This operation takes only a few seconds to perform. The small particles still clinging to the adhesive on the shoe are then removed by shot blasting; this also removes the adhesive and prepares the surface of the shoe for re-bonding. In the method used by Société Européenne de Bonding, the Redux 64 is applied only to the liner and is sprayed through a normal paint spray

These illustrations show two of the operations developed by the Société Européenne de Bonding. On the left, the old lining is being removed with a pneumatic chisel, and on the right the shoes, with their liners, are being expanded against the tension in the springs linking the two-piece steel bands, ready for hot curing





gun. In order to reduce the viscosity of the adhesive to make it suitable for spraying, approximately 10 per cent of methylated spirit is added. After a suitable time lapse, for solvent evaporation, the shoes and liners are assembled in steel bands in a manner similar to that already described in connection with the Daimler-Benz system, and the adhesive is cured in controlled temperature ovens. As the through-put is much smaller than that of a manufacturing firm, the ovens used for curing are not of the continuous feed type: instead, the oven remains closed, after being loaded, until the curing cycle is completed. After removal from the oven, the shoes are cooled before the pressure is released and the bonding inspected. The only reason for cooling them first is that it is then much easier to handle the parts than when they are removed from the oven. Inspection consists of checking that a certain amount of resin is spewed from between the liner and shoe, as evidence that the pressure has been correctly applied and to show that the colour of the cured resin is correct. It is also customary to test a certain number of bonded assemblies to destruction to ensure that the process is being correctly applied.

Société Européenne de Bonding produce a complete set of equipment the installation of which will enable any large gârage or distributor to undertake the re-bonding of brakes. This normally consists of the shot blasting cabinet, ovens, pneumatic chisels and pressure bands. It is estimated that the French bonding stations are currently assembling about 30,000 shoes per day.

New adhesive developments

The introduction of the disc brake has brought about new problems for the adhesive manufacturer, because of the different constituents from which disc pads are made. The materials used in these pads frequently differ from those used in drum brake liners, the disc brake pad being much harder and denser. These pads are in some instances chemically incompatible with the adhesives commonly used for the bonding of shoes and liners. Moreover the operating temperatures required may be much higher.

To meet these new requirements, much research has been carried out in an attempt to produce adhesives that will bond satisfactorily to the types of disc pad currently being offered. One of the difficulties encountered in this work has been the large differences in the chemical constituents of pads produced by different manufacturers. So wide are these variations that at one time it was thought that several adhesives would be necessary to cover the range of pads being offered. However, as a result of these researches, one new adhesive, known as Redux 65, has been developed, and this has shown itself capable of adhering satisfactorily to nearly all the disc pads so far submitted for test. It should be pointed out, nevertheless, that though the strength of adhesion obtained is such that, on testing, failure always occurs in the friction material, the load at which failure occurs may vary considerably. This is largely due to the variation in the cohesive strength of different pad materials some of which fail near the surface, leaving a very thin layer of the pad on the metal backing plate, whilst others fail by a deep shear in the pad material. The type of failure is usually influenced by the temperature at which the test is effected.

Redux 65 is a liquid adhesive, which is applied and cured in a manner similar to that used for Redux 64, as already described. One important difference however is that, after the adhesive has been applied, it must be dried out at a temperature of 100 deg C for a period of 10 minutes. The curing is usually carried out at a high temperature for a short time, at temperatures about 230 deg C. Disc brakes are invariably flat, and it is therefore customary to effect the cure between flat platens, in a hydraulic press or a special purpose heating jig. If the platens, which are usually electrically heated, are maintained at about 230-250 deg C and then the cold brake pad and disc inserted between them, the time required will be about 2 minutes, depending to some degree on the thickness of the backing plate. A pressure of approximately 100 lb/in^2 is necessary during this heating operation. Tests carried out, with pads from different manufacturers, with this adhesive have always resulted in a failure of the friction pad. The temperature range of these tests has varied from room temperature to 300 deg C; it is usual to find that, at room temperature, failure occurs in the pad close to the surface whilst, at temperatures of 200 deg C or more, the failure is deeper and at a lower loading. Typical results obtained with different pads are shown in the Table.

TABLE: Failing stress of different disc pads when bonded with Redux 65

Disc pad	*Temperature of test, deg C*	*Failing load, lb/in^2*	*Type of failure*
A	Room temperature	1,630	Thin shear
	200	1,630	Thin shear
	250	1,380	Thin shear
B	Room temperature	1,450	Thin shear
	200	1,190	Deep shear
	250	1,190	Deep shear
C	Room temperature	1,570	Very thin shear
	200	1,460	Thin shear
	250	790	Deep shear
D	Room temperature	1,250	Thin shear
	200	500	Deep shear
E	200	318	Deep shear
	300	227	Deep shear

Conclusion

The majority of the large European car manufacturers are now using bonding for the attachment of friction linings to brake shoes. Redux adhesives are used in most of the factories where bonding is carried out. In addition to Daimler-Benz, of Germany, whose methods have already been described, the firms of Fiat in Italy and Citroën in France also employ Redux. Citroën was probably the first European car manufacturer to rely on adhesives alone for the attachment of friction pads to backing plates for disc brakes.

Cleaning Light Alloy Die-Castings

Girling Practice for the Removal of Machining Residues and the Production of a Protective Finish

As the majority of the brake equipment made by Girling Ltd. is hydraulically operated, the component parts of this equipment require to be cleaned to a very high standard in manufacture in order to ensure consistently reliable operation in all circumstances. Light alloy components are widely used in these braking systems and various processes are employed to improve their corrosion resistances. One of these processes is the Imperial Chemical Industries' ALOCROM 1200, which is a non-electrolytic immersion treatment for producing an integral corrosion-resistant surface on aluminium and its alloys.

The coating is thin enough not to affect fine tolerances, but is sufficiently robust to protect the components, even though they are left unpainted, from corrosion caused by water thrown up by the vehicle road wheels. In addition to providing protection during storage and in service, the coating gives the component an attractive appearance. Should it require to be painted, the surface affords excellent adhesion for the paint. As the ALOCROM process is carried out at room temperature, there is no steam, spray or unpleasant fumes. Since the treatment does not require the passage of an electric current, there is no need to connect the work being processed; a time-saving and space-saving factor when dealing with large quantities of components.

To combine cleaning processes with an ALOCROM treatment, a six-stage AGI-DIP plant supplied by the Electro-Chemical Engineering Co. Ltd., of Woking, Surrey, has recently been installed by Girling Ltd. at the Company's Cwmbran factory. In this plant, aluminium alloy die-castings are treated after machining and before assembly for the removal of swarf, cutting oil, and vapour-blast residues and for the production of an ALOCROM coating. The AGI-DIP system was chosen because of its ability to deal with components of intricate design which might be damaged by excessive agitation. The system involves the parts being loaded into baskets, which are then passed automatically through six agitated immersion stages, and followed by hot air drying. Agitation is effected by means of pneumatically operated reciprocating devices which are adjustable for amplitude and frequency of stroke. The work is transported through the sequence of operations by means of an indexing beam conveyor controlled by a timing mechanism; each basket is agitated for three minutes.

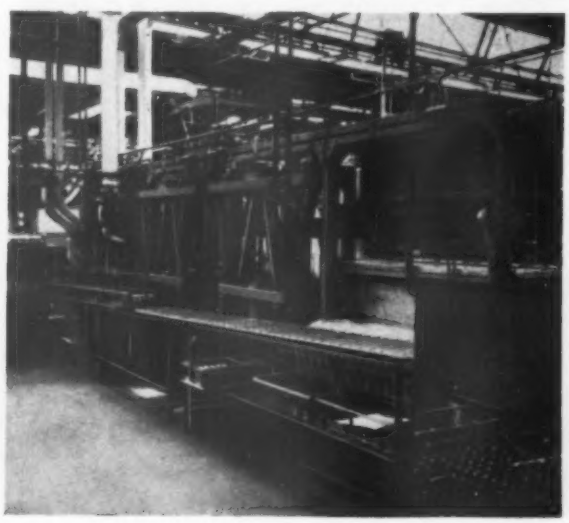
The process sequence is as follows:
(1) Emulsion pre-cleaning. (2) Alkaline cleaning at 77 deg C. (3) Cold water rinse. (4) ALOCROM treatment at room temperature. (5) Cold water rinse. (6) Hot water rinse at 98 deg C. (7) Hot air drying at 150 deg C.
The output of the plant is eighteen basket loads an hour, each basket measuring 24 in by 12 in and containing 20 lb of work. A maximum load, including basket, of 80 lb per stage may be carried. Only one operator is needed.

This type of plant is very flexible in design, and in different versions is suitable for cleaning and process treatments, including pickling and phosphating, on a great variety of components. It is operated by one transfer and three lift pneumatic cylinders working at 60 lb/in². It comprises a steel structure, on which the six tanks and the drying section stand, with the cylinders and pipe-work attached to the rear of the structure. A separate platform is supplied at each end, for automatic loading and unloading of the baskets.

The ALOCROM tank, together with its agitating frame, is constructed of stainless steel, while the remaining tanks and frames are made from mild steel and hot-dip galvanized. Basket runners on all frames are made from stainless steel. The heated tanks are thermally insulated and fitted with lip extraction. Heating is by thermostatically controlled high-pressure, hot-water coils. Baskets are also manufactured from mild steel and galvanized, each being fitted with a wire-mesh bottom and stainless steel runners. The drying section, the four sides and top of which are insulated, is constructed throughout of aluminium. Air from the blower fan, passes through an electric heater and is discharged through slotted air ducts onto the top and bottom of the baskets.

View of complete plant from the loading end, showing beam conveyor. Fume extraction hoods are in position on heated tanks

At the rear of the plant a gravity-type roller conveyor is provided for the return of work baskets to the loading station



Ceramic Coating of Wet Cylinder Liners

Method Evolved by A.E.C. Ltd. and Metallizing Equipment Co. Ltd., to Obviate Cavitation Erosion

IN diesel engines with wet liners, cavitation erosion of the exterior of the liners is a relatively familiar phenomenon. However, because of the variable factors involved, its magnitude can differ considerably between one make and another, and even between two models from the same manufacturer. At its worst, the erosion can even be sufficiently rapid to impose a limitation on the service life of the liner. It appears to be generally accepted that flexure and vibration of the unsupported portion of the liner are mainly responsible for the occurrence of cavitation. As would be expected, therefore, among the more seriously affected engines are those in which the swept volume has been increased by enlarging the bore, with a consequent reduction in the wall thickness of the liner.

Although cavitation erosion has never reached critical proportions on the A.E.C. engines, its incidence has, nevertheless, been enough to warrant the investigation of corrective techniques by the company. Brief reference was made in the November 1958 issue of *Automobile Engineer* to the fact that a ceramic coating was applied to the exterior of the liners on the two latest engines—the 9·6-litre A590 and the 11·3-litre A690. At that time, it was not possible to give a more detailed description of the process employed, because certain aspects were still experimental. These aspects have since been finalized, and full information on the plant and its operation has recently been released by A.E.C. Ltd. All wet cylinder liners manufactured by the company are now treated in this way.

When the decision was taken, about two years ago, to adopt surface treatment to counter the erosion of the liners, attention was first directed towards the electrodeposition of chromium or copper. The first objection to plating was the necessity of installing new and relatively costly plant, since the existing equipment at the Southall factory was of insufficient capacity. Secondly, the time cycle for plating and additional handling would be so lengthy as to necessitate a large float of liners, in order that engine production should not be held up.

For successful ceramic coating, the liners must be quite free of grease and oil, and have a suitable keying surface. A Spencer Halstead blast machine is employed to treat the exterior of the liners before spraying

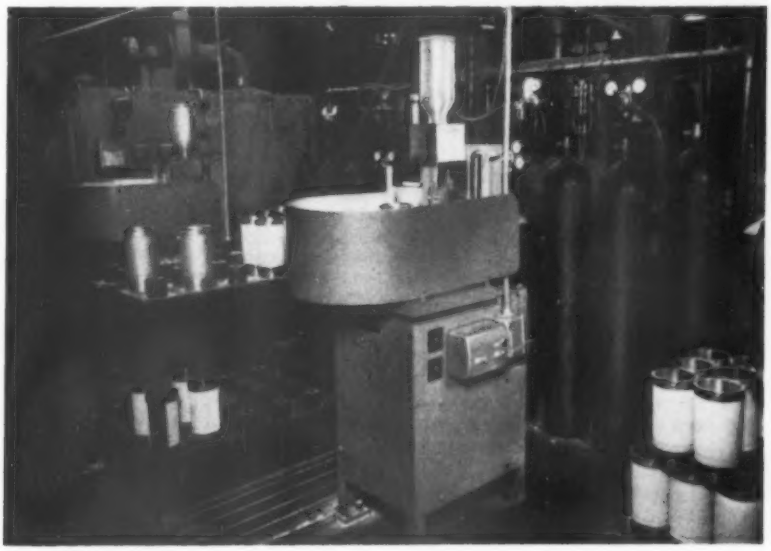
The possibilities of metal spraying—the flame deposition of powdered metal—were then investigated, with particular reference to copper. Consultations with Metallizing Equipment Co. Ltd., 60 Victoria Street, London, S.W.1, indicated that such a coating would be adequately free from porosity, and could be produced on a suitable time cycle. However, the project was not taken far because it was felt that a coating of this type might not be strong enough to withstand the cavitational conditions. It was at this stage, in September 1957, that Metallizing Equipment Co. suggested spraying aluminium oxide in powder form, by means of their ThermoSpray process, which at that time was interesting the Admiralty, Brush Group, Rolls-Royce, and Davey Paxman and Co.

As is well known, aluminium oxide, or alumina, as it is commonly called, has largely ousted porcelain in many applications, including the insulators of sparking plugs. It has a tensile strength of over 17,000 lb/in^2, a compressive strength above 300,000 lb/in^2, and is of the same hardness as sapphire, namely 9, as compared with the 10 of diamond, on the Moh scale. In addition, even a thin coating has been found very effective as a thermal barrier to prevent the oxidation of metals at temperatures of up to 2,000 deg C.

Since ceramic coating appeared to offer considerable possibilities, A.E.C. decided to investigate its effectiveness, so four sample liners were submitted for treatment by the ThermoSpray process. In view of the desirability of obtaining maximum toughness of the coating, a small proportion of titanium was added by Metallizing Equipment Co. to the alumina powder used. The production coatings on the liners have, in fact, this same additive. Two of the experimental liners were submitted to laboratory tests at Southall and the others were fitted to an engine for field testing.

The results of these tests were satisfactory, and the next stage was the production of a pilot batch of 100 liners, in order to gain knowledge of the technique of coating and to ascertain the degree of consistency obtainable. Because Metallizing Equipment Co. does not undertake contracting of that type, the work was passed to another firm. The liners of this batch were fitted to production engines, and shortly afterwards work was put in hand at Southall for building the plant to coat the liners there. Since this plant was not unduly costly, it was considered preferable from both the economic and technical viewpoints to bring manufacture within the factory. Production of the coated liners actually began in August last year, and an output rate of 900 liners a week can be attained if necessary, though the actual figure naturally depends on the immediate requirements for engines on the production line and for spares.

Two prerequisites for good adhesion of the coating are absolute freedom from grease or oil, and a suitable keying surface. They have been met by the installation of a Spencer Halstead abrasive blast machine, through which the liners are passed after their exteriors have been finished and the initial boring has been effected. To economize on handling time and labour, this plant and the two for the coating process are grouped close to the machining section. A coarse, carbon silicide grit is employed for the blasting, which is carried out on a four-minute cycle to match the output of the machines that apply the ceramic coating.



General view of the two machines that coat the liners by the ThermoSpray process. In the background may be seen the abrasive blast machine, and on the right of the nearer spraying plant is a stack of coated liners awaiting finishing of the bores

It was mentioned in the preceding paragraph that there are two of these machines. This duplication was necessary, in view of the time cycle of the process, to meet production demands for the wet liners. The basic requirements for the machines, in terms of feeds and speeds, gas consumption and the siting of the gun relative to the work, were laid down by Metallizing Equipment Co., but the actual design and construction were done by A.E.C. Ltd.

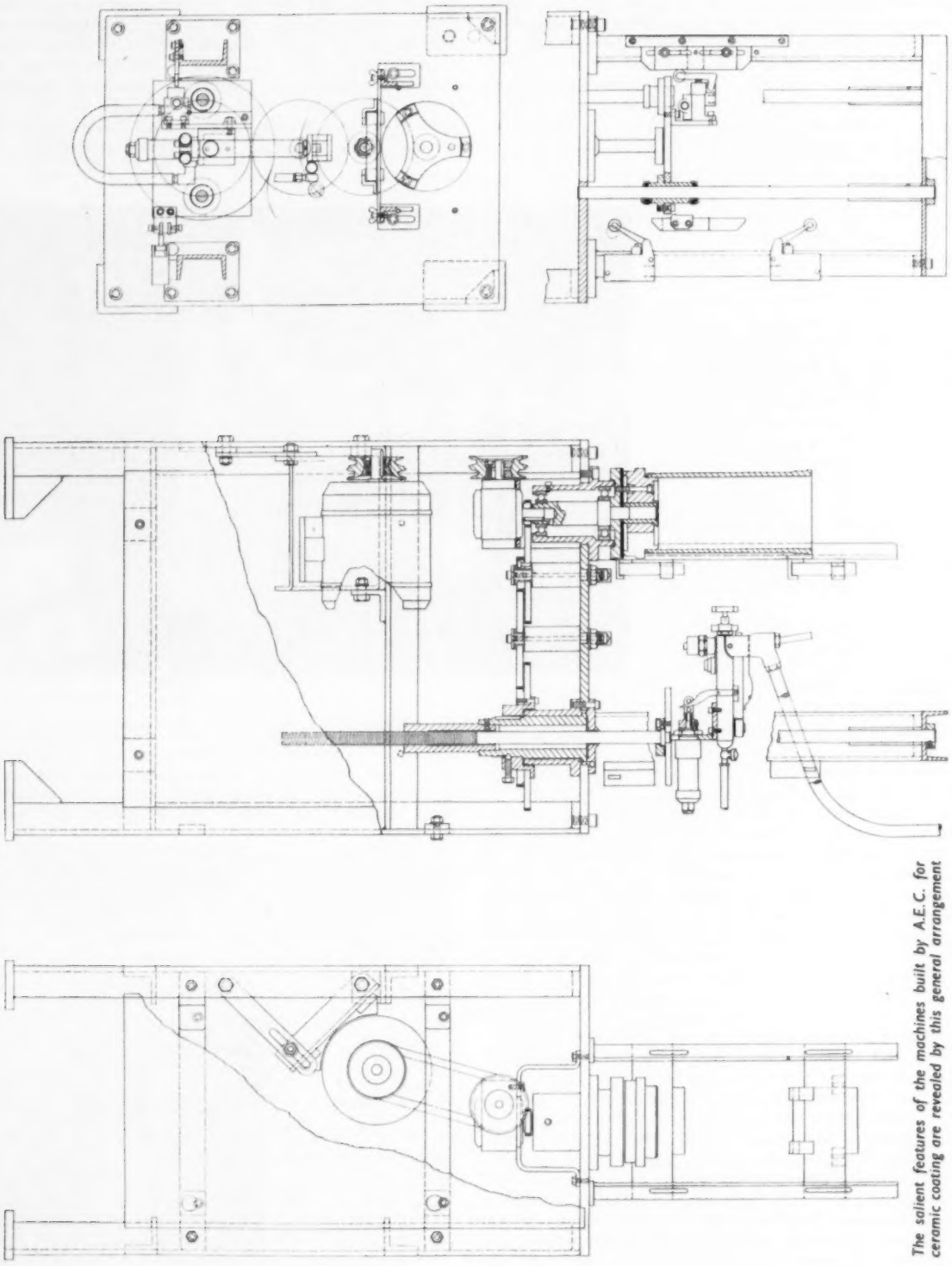
Each machine comprises three main assemblies: the electric motor, the turntable carrying the liner, and the vertical traverse mechanism for the gun that sprays the alumina. These assemblies are housed in a simple structure consisting mainly of steel angle-section and sheet. The ten gas cylinders—oxygen and acetylene—for the guns are stowed in line behind the two machines, and the piping is laid out for easy switching from one to another.

The ½ h.p. motor, sited below the turntable, drives a small David Brown worm gearbox, by means of a vee-belt and pulleys. Adjustment of the motor position can be effected, for belt tensioning, and each pulley is of the double type to give a choice of two input speeds. The lower end of the liner slips over a spigot on the turntable, which is driven directly by the output shaft of the gearbox. Two ball bearings carry the turntable shaft; a modification not shown in the accompanying reproduction of a drawing of the machine is the fitting of a water jacketed housing for the bearings. The jacket was found necessary because of the considerable conduction of heat from the liner during spraying, and it is fed from the mains supply. Further cooling is effected by three scallops cut in the turntable spigot, which permit an induced draught of air, under the influence of convection, to flow up the bore of the liner.

Guide rails support the gun unit, which is raised or lowered by means of a screw jack having 8 turns per inch. The nut of the jack is driven from the gearbox output shaft by a train of four gears. Two further speed variations are provided in this gear train, by alternative second intermediate gears; when the larger of these is in use, the spindle on which it is mounted is moved to an offset position, for correct meshing. To give the gun the appropriate travel for the particular liner being coated, cam operated limit switches with vertical adjustment are employed; they stop the motor when the limit of movement is reached in either direction. The motor is of the reversible type and the circuit is so arranged that, after one of the limit switches has been tripped, actuation of the starting button automatically restarts the motor in the reverse direction of rotation.

A minimum deposit efficiency of 97 per cent is claimed for the ThermoSpray gun, an interesting feature of which is the cooling ring mounted ahead of the nozzle. This cooling ring serves to keep down the temperature of the workpiece, and it is employed in any application where the heat transfer is large. The ring contains an internal passage

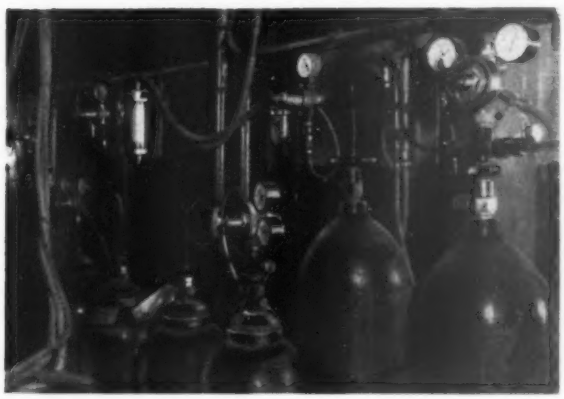
This photograph shows the roughening of the liner surface, which is produced by the blast machine whose nozzle is on the left-hand side



The salient features of the machines built by A.E.C. for ceramic coating are revealed by this general arrangement

that is connected to the compressed air supply in the shop; facing forward from the passage is a series of holes that direct the air on to the perimeter of the area covered by the spray from the gun. Another noteworthy feature is that, for ceramic coating, the ThermoSpray gun and its feed hopper are each fitted with an electrically operated vibrator. Without these vibrators, there would be a tendency for the powder to pack. The irregular flow resulting from such packing would manifest itself in an uneven thickness of the coating.

To ensure that the coating covers only the appropriate section of the liner, upper and lower masking pieces are mounted on a vertical framework between the gun and the turntable. As in the case of the limit switches already mentioned, provision is made for vertical adjustment in the mounting of the masks. Because of the high temperature of the spray, it has been found necessary to use $\frac{3}{8}$ in steel plate for the masks; thinner material, as originally tried, was too rapidly burnt away. Adjustable cams on the gun-carrying structure are aligned with the masks so that their



followers cut off the air supply to the nose ring at the extremes of travel. A pneumatic switch in the air circuit automatically shuts off the powder feed at the same time.

The nozzle of the gun is mounted approximately $3\frac{1}{2}$ in from the surface of the liner—the distance is not critical within $\frac{1}{2}$ in—and the gauged thickness of the coating is 0·006 to 0·010 in, the average being about 0·007 in. A thicker coating could readily be applied, but is not regarded as necessary or desirable. The minimum nominal thickness quoted is ample to ensure a complete absence of porosity; tests have indicated that the porosity is under 6 per cent at the surface and becomes nil before half-depth is reached. Increasing the thickness would not, therefore, result in any better protection of the liner. Also, since the coefficient of linear expansion of aluminium oxide is approximately half that of iron, a thin coating would, because of its greater elasticity, be less prone to cracking through stresses arising from differential expansion. Finally, a thinner coating is obviously less expensive and quicker to apply than one of greater thickness, a point of some importance in view of the desirability of keeping the cost per liner as low as possible.

Mention has already been made of the importance of consistency in the powder flow. Serious packing results from any trace of dampness in the powder and, to ensure the exclusion of moisture, it is stored in bulk at Southall in a warm, dry atmosphere. From the store, it is decanted as required into screw-top tins which, as a further precaution, are normally placed by the operators within liners that are still hot after completion of their coating.

Left: Some of the ten cylinders of oxygen and acetylene for the spray machines, showing the piping layout. Above: In this view can be seen the liner turntable and the guide-rail mounting of the ThermoSpray gun

Even with thoroughly dry powder, however, some difficulty was experienced in the early stages through its clogging in the feed system. It can be seen that the drawing reproduced shows a mounting for the hopper on the main frame, with a relatively long tube leading directly from it to the gun. This arrangement was the cause of the trouble, and it was soon altered to that shown in one of the photographs. The main hopper now forms part of the traversing unit, and a short, nearly vertical tube leads from it to a small, secondary hopper on the gun. This secondary hopper is of the type used when such a gun is operated by hand. No further difficulty has been experienced since these modifications were made.

For obvious reasons, A.E.C. is unwilling to disclose the actual additional cost of the ThermoSpray process per liner. In any case, this is dependent on the coated area, which varies by 80 per cent between the smallest and the largest liners treated. It may be revealed however, that approximately 1·3 ounces of the ceramic material is used on each liner for the familiar 6·75-litre A410 and 7·685-litre A470 engines. Gas consumption naturally varies with the rate of output, but averages about three cylinders each of oxygen and acetylene during a normal working day.

Push Button Pencils

A PUSH button pencil, called the Koh-i-Noor Versatil No. 828, has been introduced for use in drawing offices. It has a metal body and a knurled finger grip, and is available in six colours. Leads ranging from 2B to 9H can be supplied. A slightly smaller and cheaper version, called the Koh-i-Noor Baby Versatil 829, is also offered. This model is designed mainly for students. Leads from B to 9H can be supplied, and the pencil has a plastics body, also available in six colours. Yet another version is the Super Versatil 827. This pencil, which was first introduced about two years ago, takes leads from 6B to 9H.

The firm also announce a new pack for their range of drawing leads in the degrees ranging from 6B to 9H, and in the colours red, blue, green, yellow, magenta and violet. It consists of a neat plastics box with a slide-lid and contains two leads. The manufacturers feel that this will be welcomed by purchasers whose requirements are small.

COIL SPRING PERFORMANCE

Stresses Due to Impact Loading in Helical Compression Springs

S. C. Gross, Dipl. Ing.

According to earlier published work the maximum total stress of a dynamically loaded spring is the algebraic sum of the separately computed static and dynamic stresses. This later study shows that this concept is never correct and that the maximum total stress is equal to twice the dynamic stress.

SUDDEN displacement of the free end of a helical spring takes a certain time to affect all coils. The disturbance caused by the displacement travels along the spring axis towards the fixed end with a velocity u which depends on the dimensions and the material of the spring. As the spring is gradually affected by the disturbance, the distribution of displacement, load and stress cannot be uniform. Therefore the stress must locally be larger than the stress due to a static compression of equal magnitude.

Impact loading can be represented by the action of a wedge W, Fig. 1, of maximum height δ and length L_w, moving perpendicularly to the axis of the spring S_p with a constant velocity v_w such that the compression of the spring is completed in T sec. Fig. 1 shows the state of affairs t sec after compression has begun. Edge A' of the wedge has travelled $z=v_w t$ after passing the spring axis, and has displaced the free end of the spring by y. In the meantime, the disturbance caused by edge A' has covered the distance $L-x=ut$ on the spring axis and reached point A. The ordinates of the curve AP represent the displacements of the various points of the spring at time t. Obviously, this curve, the wave, differs from the contour $A'P'$ of the wedge only by the lengths of the abscissae, the ratio of which is u/v_w. If, however, the wedge is moved with velocity u and if its abscissae are altered in the ratio u/v_w, compression δ will again be completed in T sec, and the contour of this new wedge is identical with the wave of length λ.

The wave velocity with respect to the spring axis is given by:

$$u=L\sqrt{\frac{sg}{\rho Al}} \tag{1}$$

where

- L instantaneous length of spring
- s stiffness of spring
- g gravitational acceleration
- ρ density of spring material
- A cross-sectional area of spring wire or bar
- l length of active wire or bar

The units are in, lb and sec throughout.

Because u varies with L, which is continuously changing, it is often more convenient to use the constant wave velocity c with respect to the length of the spring wire and, accordingly, to relate the wave to the wire. In the wave diagram, Fig. 1, L has to be replaced by l, and x has to be measured on the helix. Velocity c is obtained by multiplying (1) by l/L, that is:

$$c=\sqrt{\frac{sgl}{\rho A}} \tag{2}$$

Since s denotes the stiffness of a spring with length of wire l, the stiffness per unit of length of sl, and that of an element of wire dx is $s'=sl/dx$. Therefore, the force that produces the deflection dy of this element, in the direction of the spring axis, is given by:

$$P=s'dy=sl\frac{dy}{dx} \tag{3}$$

In this equation, the differential quotient dy/dx is the slope of the tangent to the modified wedge at a point with ordinate y and abscissa x and, consequently, that of the tangent to the corresponding point of the wave. Therefore, P is the force acting in the wire element dx at the passage of the point of the wave under consideration.

By multiplying (3) by dt/dt:

$$P=\frac{sl(dy/dt)}{dx/dt} \tag{3a}$$

Now, dy/dt is the velocity v with which the element is deflected, and dx/dt the constant wave velocity c with respect to the spring wire so that (3) may be written:

$$P=sl\frac{v}{c} \tag{3b}$$

The maximum force P_m is proportional to the maximum velocity v_m imparted to the free end of the spring when the steepest part of the wave enters the spring. The stress which P or P_m sets up in the spring can be calculated by means of the conventional formula for the corrected stress which, if (2) and (3b) are considered, takes the form:

$$q=2v\sqrt{\frac{\rho G}{2g\nu_w}}$$

where ν_w is the weight efficiency of the spring, that is, the ratio of the energy stored in a spring with length of active wire l to that in a straight bar of equal cross-section and length, if the stress has the same magnitude in both cases—with a round bar, $\sqrt{1/\nu_w}=K=$Wahl's correction factor. If v is in ft/sec and g in ft/sec², and if q is given the suffix v, this formula becomes:

$$q_v=v\sqrt{\frac{24\rho G}{g\nu_w}}$$

and is now identical with that given on page 4 of Sheet DE4,

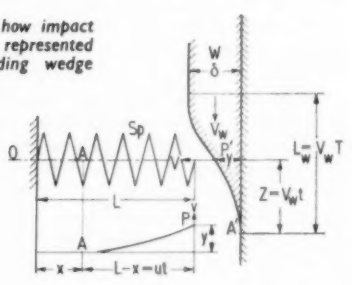


Fig. 1. Diagram showing how impact loading of a spring can be represented by the action of a sliding wedge

of the Design Data Sheets for the Design of Helical Compression Springs, published by the Ministry of Supply. According to this data sheet, the maximum total stress in an impact-loaded spring is the sum of the stress corresponding to the maximum deflections that the spring undergoes and the stress q_v corresponding to v_m. The purpose of the following investigation is to prove that this statement is not correct.

Under the conditions already mentioned, when the head A of the wave has reached the free end O of the spring and travels on, the wave is reflected as any other compressive, longitudinal, or transversal wave is reflected by a rigid wall. For the purpose of analysis, it is assumed that the wave enters the wall and that its part which is still outside interferes with an identical but negative wave emerging from the wall with equal velocity c and phase x_o. In Fig. 2, length $AO=X_o$ of the rising part ACB of the original, positive, wave has entered the wall and length $A'O=x_o$ has emerged. Considering two points P_1 and P_2 on the original and the negative wave, respectively, with abscissa x_r and ordinates y_1 and y_2, and subtracting y_2 from y_1, the ordinate $y_r=y_1-y_2$ of point P_r on the reflected wave and, thus, its whole contour OP_rA'', is obtained. Since the ordinates of both waves are always inversely equal at the reflecting wall, the reflected wave has its origin in O. For reference at a later stage of this investigation, it is necessary to keep in mind that, from the moment of entering the spring at its free end to the phase x_o of the reflection at the fixed end, head A of the original wave has covered the distance $l+x_o$ in the spring.

If $y=f(x)$ is the equation of the wave, the ordinate of any point y_r on the reflected wave is given by:

$$y_r=y_1-y_2=f(x_o+x_r)-f(x_o-x_r) \tag{4}$$

In order to find the slope of the reflected wave at P_r, differentiate (4) with respect to x_r:

$$\frac{\partial y_r}{\partial x_r}=\frac{\partial}{\partial x_r}f(x_o+x_r)+\frac{\partial}{\partial x_r}f(x_o-x_r) \tag{5}$$

Because the two items of the right hand term are the slopes of the tangents to the original and the negative waves at P_1 and P_2, respectively, the steepness of the reflected wave equals the sum of the steepnesses of the two waves. At the fixed end of the spring, where $x_r=0$:

$$\frac{\partial y_r}{\partial x_r}=2\frac{\partial}{\partial x_r}f(x_o)=2\frac{dy}{dx_{(x=x_o)}} \tag{5a}$$

That is, at the fixed end, the reflected wave is twice as steep as the original one. Consequently, the maximum dynamic spring load occurs at the fixed end and is given by:

$$\left[cf\,(3),\,(3a)\text{ and }(3b)\right]\quad P_m=2sl\left(\frac{dy}{dx}\right)_{max}=\frac{2slv_m}{c} \tag{6}$$

Since the maximum static load, corresponding to compression δ is given by $P_{st}=s\delta$:

$$P_m/P_{st}=2\frac{l}{\delta}\left(\frac{dy}{dx}\right)_{max}=2\frac{l}{\delta}\frac{v_m}{c} \tag{7}$$

The shape of the reflected wave at the various phases of reflection can conveniently be found by a method demonstrated clearly and in detail by the following example.

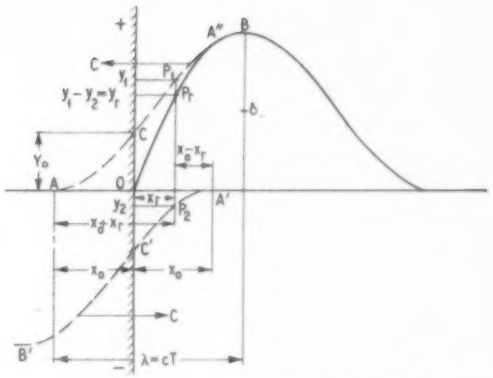


Fig. 2. When the head A of the wave has reached the free end O of the spring and travels on, it is reflected as are other types of waves

The original wave, as shown in Fig. 2, has the equation:

$$y=\tfrac{1}{2}\delta(1-cos\tfrac{x}{\lambda}\pi)\text{ or }y/\delta=\tfrac{1}{2}(1-cos\tfrac{x}{\lambda}\pi)$$

Divide the wave length λ into an adequate number of equal parts, for example twelve, and compute or, in the case of an empirically given wave, measure the ordinates y/δ for $x/\lambda=0$, 1/12, 2/12, etc. By writing these ordinates on a strip of paper from left to right, a positive scale is obtained. A negative scale is made by writing the ordinates from right to left on another strip. Put the negative scale below the positive one, so that the y/δ value for $x/\lambda=0$ on the former is below the always identical y/δ value for $x/\lambda=0$ on the latter. This position of the scales corresponds to the arrival of head A of the original wave at the fixed end O. From $x/\lambda=0$ to the right, are the y/δ values of the original wave, which is still entirely outside the reflecting wall, and to the left, are the y/δ values of the negative wave, which has not yet emerged. Now, move both scales by 1/12, 2/12, etc. in opposite directions with respect to a stationary vertical line representing the fixed end O, the positive scale to the left and the negative to the right, so that always identical y/δ values meet on the line of reference. By subtracting the values on the negative scale from those on the positive one, both on the right of the vertical, we obtain y_r/δ. The position of the scales and the result for the phase $x_o=5/12\lambda$ is shown in Fig. 3.

The same method can be used for finding $\partial y_r/\partial x_r$. By differentiating the equation of the original wave, or by graphical differentiation of an empirically given wave we obtain

$$\frac{dy}{dx}=\frac{\pi}{2}\frac{\delta}{\lambda}\,sin\,\frac{x}{\lambda}\pi$$

and the scale in Table I.

To obtain $\partial y_r/\partial x_r$, corresponding values on upper and lower scale have to be added to each other, *cf* (5). If, however, it is only required to find $(dy/dx)_{max}$, in order to calculate

Fig. 3. Values of y_r/δ obtained, for the phase $x_o=5/12\lambda$, by using the positive and negative scales as described on this page

x/λ	0	1/12	2/12	3/12	4/12	5/12	6/12	7/12	8/12	9/12	10/12	11/12	12/12
y/δ	0	0·017	0·067	0·147	0·250	0·371	0·500	0·629	0·750	0·854	0·933	0·983	1·000

	0·854	0·750	0·629	0·500	0·371	0·250	0·147	0·067	0·017	0			
y_r/δ	=0					0·250	0·482	0·683	0·837	0·933	0·983	1·000	
	Point 0									Point A''		Point B	

x/λ	0	1/12	2/12	3/12	4/12	5/12	6/12	7/12	8/12	9/12	10/12	11/12	12/12
$\sin\frac{x\pi}{\lambda}$	0	0·259	0·500	0·707	0·866	0·966	1·000	0·966	0·866	0·707	0·500	0·259	0

Table I: SHOWING THAT THE MAXIMUM VALUE OF $\sin\frac{x}{\lambda}\pi$ IS EQUAL TO UNITY, AND THAT P_m WILL OCCUR AT THE PHASE $x_o=\frac{1}{2}\lambda$

P_m and P_m/P_{st} from (6) and (7), the lower scale is unnecessary. The scale in Fig. 4 shows that $sin\frac{x}{\lambda}\pi_{\ max}$ equals unity, and that P_m occurs at the phase $x_o=\frac{1}{2}\lambda$. Therefore

$$(dy/dx)_{max}=\frac{\pi}{2}\frac{\delta}{\lambda}$$

and, by substituting this term in (6) and (7),

$$P_m=\pi s\frac{l}{\lambda}\delta=\pi s\frac{l}{cT}\delta \text{ and } P_m/P_{st}=\pi\frac{l}{\lambda}=\pi\frac{l}{cT} \qquad (7a)$$

It is apparent that P_m is proportional to the ratio $l/\lambda=n$ or to l/T. This fact tends to indicate that P_m decreases indefinitely as λ or T are increased so that, in the limit, $P_m\to 0$ as $\lambda\to\infty$ or $T\to\infty$, that is, in the case of static loading; it also tends to indicate that P_m is actually an additional load to be added to the maximum static load P_{st}. However, it will be proved that this is not true because, below a certain value of n, the numerical factor, in this case π, increases and, what is more, increases at a higher rate than the ratio n is decreased, so that the product of the numerical factor and n approaches unity as limiting value if $n\to 0$.

If the wave is twice as long as the wire, $n=l/\lambda=\frac{1}{2}$, the head of the wave has travelled from the free to the fixed end and back and covered the distance $2l$ when compression δ is completed at time T. Apparently, equation (4) fully covers this event and all those corresponding to greater values of n.

If, however, $n<\frac{1}{2}$, the negative wave is reflected by the free end before time T; that is, an imaginary positive wave emerges from the free end with velocity c and, in travelling from right to left, interferes with the negative and original wave.

If $n<1/3$, the head of the new positive wave will have reached the fixed end and be reflected before time T. Thus, if n is decreased, more and more reflections take place, alternately at one end and at the other.

The further reflections can be accounted for by a completed version of (4) which is:

$$y_r=f(x_o+x_r)_{x_o>0}-f(x_o-x_r)_{x_o>0}+f(x_o+x_r-2n\lambda)_{x_o>n\lambda}$$
$$-f(x_o-x_r-2n\lambda)_{x_o>2n\lambda}+f(x_o+x_r-4n\lambda)_{x_o>3n\lambda} \qquad (8)$$

The suffixes to the individual terms of this summation indicate the phase at which they become effective, that is, have a positive argument. For example, if $x_o=n\lambda=l$ is put in the argument of the second positive term and the head of the first negative wave has the abscissa $x_r=n\lambda$, at its arrival at the free end, $x_o+x_r-2n\lambda=0$; that is, the term becomes effective when $x_o>n\lambda$.

By combining the positive and negative terms of (8):

$$y_r=f(x_o+x_r)_{x_o>0}+f(x_o+x_r-2n\lambda)_{x_o>n\lambda}$$
$$+f(x_o+x_r-4n\lambda)_{x_o>3n\lambda}+\ \ldots\ .-$$
$$-\Big[f(x_o-x_r)_{x_o>0}+f(x_o-x_r-2n\lambda)_{x_o>2n\lambda}+$$
$$f(x_o-x_r-4n\lambda)_{x_o>4n\lambda}+\ \ldots\ .\Big] \qquad (8a)$$

This shows that the reflected wave is the resultant of the superposition of two identical trains of waves, which travel in opposite directions.

By applying this theorem to the example for the case $n=1/6$, it is possible to obtain the positive summation:

$$y/\delta=\tfrac{1}{3}\Big[(1-cos\frac{x}{\lambda}\pi)+(1-cos\frac{x-1/3\lambda}{\lambda}\pi)+$$
$$(1-cos\frac{x-2/3\lambda}{\lambda}\pi)+(1-cos\frac{x-3/3\lambda}{\lambda}\pi\Big]$$

and the positive scale shown in Table II.

Differentiating the equation for y gives:

$$dy/dx=\frac{\pi}{2}\delta/\lambda\Big[sin\frac{x}{\lambda}\pi+sin\frac{x-1/3\lambda}{\lambda}\pi+sin\frac{x-2/3\lambda}{\lambda}\pi+$$
$$sin\frac{x-\lambda}{\lambda}\pi\Big]=\frac{\pi}{2}\delta/\lambda\ \Sigma\ sin$$

and, for $\Sigma\ sin$, the scale is as shown in Table III.

The maximum value of $\Sigma\ sin$ is 2·0 and occurs at the phase $x_o=5/6$. Now,

$$(dy/dx)_{max}=\frac{\pi}{2}\delta/\lambda\ \Sigma\ sin_{max}=\pi\delta/\lambda$$

and, therefore,

$$P_m/P_{st}=2l/\delta\ (dy/dx)_{max}=2\pi l/\lambda=2\pi n=\frac{\pi}{3}=1\cdot048$$

From (7a), however, the value obtained would have been: $P_m/P_{st}=\pi l/\lambda=\pi n=\pi/6=0\cdot524$.

In this example, the last term in each summation could

Table II: THE REFLECTED WAVE RESULTING FROM THE SUPERPOSITION OF TWO IDENTICAL TRAINS OF WAVES TRAVELLING IN OPPOSITE DIRECTIONS

x/λ	0	1/12	1/6	1/4	1/3	5/12	1/2	7/12	2/3	3/4	5/6	11/12	1
	0	0·017	0·067	0·147	0·250 0	0·371 0·017	0·500 0·067	0·629 0·147	0·750 0·250 0	0·854 0·371 0·017	0·933 0·500 0·067	0·983 0·629 0·147	1·0 0·750 0·250 0
y/δ	0	0·017	0·067	0·147	0·250	0·388	0·567	0·776	1·0	1·242	1·500	1·759	2·0

Table III: FROM THIS TABLE IT CAN BE SEEN THAT THE MAXIMUM VALUE OF ΣSIN IS 2·0 AND OCCURS AT THE PHASE $x_o=5/6$

x/λ	0	1/12	1/6	1/4	1/3	5/12	1/2	7/12	2/3	3/4	5/6	11/12	1
	0	0·259	0·500	0·707	0·866 0	0·966 0·259	1·0 0·500	0·966 0·707	0·866 0·866 0	0·707 0·966 0·259	0·500 1·000 0·500	0·259 0·966 0·707	0 0·866 0·866 0
$\frac{1}{3}\Sigma$ sin	0	0·259	0·500	0·707	0·866	1·225	1·500	1·673	1·732	1·932	2·0	1·932	1·732

have been omitted because both equal zero when they become effective at the phase $x_o = \lambda$, but it is not permissible to neglect the last term in the summation of dy/dx if, in a similar case, this term has a finite value.

To prove that in all instances $P_m/P_{st} > 1$, assume that the spring is compressed with a very small constant velocity v_c of its free end, so that it takes a very long time T to complete compression δ. After loading has started, a wave of constant velocity v_c is travelling from the free to the fixed end, and when it has arrived there, after $t = l/c$ sec, the whole of the spring is moving with velocity v_c. After $2t = 2l/c$ sec, the head of the wave will arrive at the free end, and the whole of the spring will be at rest. Suppose this happened a great number of times m during the T sec of loading. Then:

$$T = m\frac{2l}{c} = \frac{\delta}{v_c} \qquad (9)$$

On the other hand, the velocity v_m, which occurs in (6) and (7) and, now, has also to be supposed constant, can be equated to mv_c, so that $v_c = v_m/m$. Then, from (9):

$$m\frac{2l}{c} = \frac{m\delta}{m} \qquad \text{or } v_m = \frac{\delta c}{2l}$$

and from (6) and (7)

$$P_m = s\delta = P_{st} \qquad \text{and } P_m/P_{st} = 1.$$

This investigation shows that the equations (6) and (7) in the forms which contain v_m must not be used indiscriminately, and that it is only possible to be sure of correct results if $T \leq 2l/c$, or $n \geq \frac{1}{2}$; although, in some cases, T may slightly exceed this limit.

If $n < \frac{1}{2}$, P_m/P_{st} does not approach unity in a continuous curve, but does so by leaps. Consider a wave with the equation

$$y = (\sqrt{2}-1)\delta\,[1 - \sqrt{2}\cos(3/4x/\lambda + 1/4)\pi]$$

which might occur in the recuperator spring of a gun, and whose shape is reproduced in the upper left-hand corner of Fig. 4. This illustration shows P_m/P_{st} as a function of n.

Down to $n = 0{\cdot}437$, P_m/P_{st} follows a straight line which, when produced, intersects the P_m/P_{st} axis at $P_m/P_{st} = 0$. If n is decreased, P_m/P_{st} rises from 1·206 to a maximum value of about 1·31, and then decreases to 1·126 for $n = 1/4$; next, it follows a further rise to about 1·19 and a drop to 1·110 for $n = 1/6$, and so on. The height and range of the leaps diminish with decreasing values of n, and there can be no doubt that $P_m/P_{st} \to 1$ if $n \to 0$.

The P_m/P_{st} values corresponding to the points of return, with abscissae $n = 0{\cdot}437$, 1/4, 1/6, 1/8, etc., are attained at two phases of reflection. From $n = 1/4$ downwards, these points always occur if λ is increased by $2l$, that is, if a further reflection takes place.

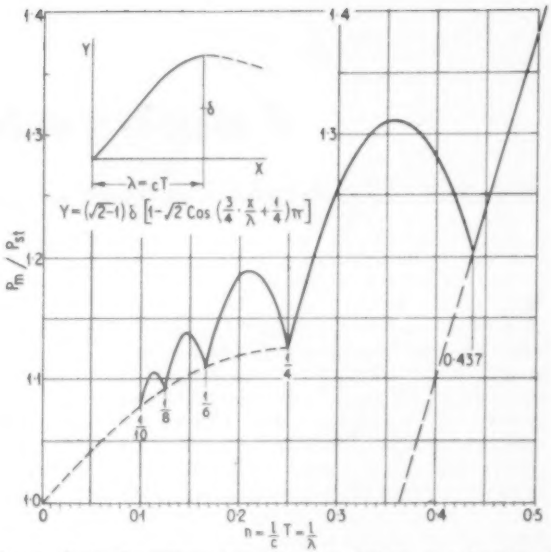


Fig. 4. Ratio of maximum dynamic: static loading, as a function of the inverse wavelength. The shape of the wave is shown inset

Conclusions

Maximum dynamic load, and stress, due to the velocity of loading is twice that from Data Sheet DE4. The formulae in this publication only hold if the length of the compression wave is shorter than twice the length of the active spring wire. On the other hand, this dynamic load is not an additional load to be added to the static load. It includes the static load corresponding to the maximum displacement of the free end of the spring during dynamic loading, but not any initial load. To obtain the total load, it is necessary to add the initial to the dynamic load. The load thus obtained, however, has no meaning if it exceeds the static load that closes the coils. On the other hand, the load due to maximum acceleration as mentioned in both publications is really an additional load which even has to be added to the static load required to close the coils of the spring.

Obstructions in Pipe-Lines

SAFETY precautions for workers who may be exposed to ionizing radiations arising from the inclusion of radioactive markers in Go-devil devices for the detection of obstructions in pipe-lines are set out in a new booklet "Radioactive Markers in Go-devils; Safety Precautions", recently published by the Ministry of Labour and National Service. Go-devils consist of tightly fitted rubber washers, scraper vanes or wire brushes mounted on a central core. They are pushed through the pipe-lines by means of water or compressed air, and they sometimes get jammed against an obstruction. Their location can be an expensive procedure, but the inclusion of a radioactive marker in the central core, which is exposed on contact with an obstruction, will overcome the difficulty because the gamma rays given off by the radioactive material can be readily traced, even if the pipes are underground.

Because of the use of radioactive substances in these devices, the observance of strict precautions is necessary to avoid hazards. A Code of Practice for persons engaged in such operations is set out in the new publication. This is an illustrated booklet and is number 7 of a series, dealing with safety, health and welfare in industry. The series is now being prepared by the Ministry of Labour and National Service. This particular booklet is available, price 1s, from H.M. Stationery Office, Atlantic House, Holborn Viaduct, London, E.C.1.

Paint Stripper for Plastics Panels

GLASS fibre reinforced panels are increasingly being adopted for vehicle bodywork. During renovation or repair it has been found exceptionally difficult to remove old paint from such panels. The firm of J. Manger and Son, Ltd., 57d Kingsland High Street, London, E.8, has recently developed a stripper specifically for this purpose. Known as Quickerstrip 104/2, this composition is fairly viscous for easy control on vertical surfaces and can be rinsed off with water. Supplies are now available from the usual sources.

Combustion Control

Reduction of Knock Intensity in Spark Ignition Engines by the Use of Precombustion Chambers

LEON GROSS-GRONOMSKI, Dr. Ing. *

THIS is a paper concerning the problem of reducing the knock intensity of spark-ignition engines, to permit the use of an increased compression ratio with a standard fuel, or, alternatively, the use of poor-quality fuel with a low octane number on an existing engine without alteration to the compression ratio. The author has been able to improve a petrol engine in this respect by the use of a precombustion chamber surrounding the sparking plug. The precombustion chamber communicates with the cylinder via holes of special design drilled into the chamber wall.

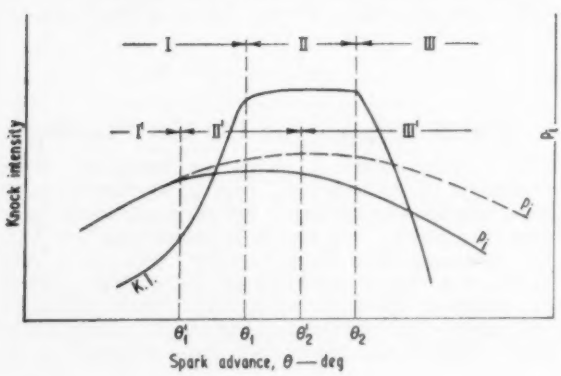
Obviously, the degree of improvement of the knock intensity obtained in this way in any engine depends on suitable design of the precombustion chamber for that particular type of engine. Parameters influencing precombustion chamber performance are: the volume of the chamber, the size, number, shape, orientation, and geometrical distribution of the holes drilled into its walls, and the heat-conduction properties of the chamber.

The paper deals briefly with general considerations, a description of tests carried out with precombustion chambers, a survey of test results with a number of different designs, and an analysis of test results. No other modification of the engine was necessary when precombustion chambers of the author's design were used. From data given in the paper, conclusions may also be reached on the influence that the introduction of a precombustion chamber has on engine output and efficiency. The results obtained from the tests show that, with a properly designed precombustion chamber, the knock intensity of the engine is significantly reduced.

Influence of ignition timing on knock intensity

To prevent knocking in spark-ignition engines, the spark-advance angle is customarily restricted. However, in the course of experiments conducted by the author, it was found that, if the influence of ignition timing on knock intensity is tested over a wide range of spark-advance angles, then, in a certain range, a reduction of the angle may produce an increase in the knock intensity. Within an intermediate range of spark-advance angles, a change in the angle does not, for all practical purposes, affect the knock intensity. These phenomena are illustrated in Fig. 1.

Fig. 1. Iso-compression-ratio curves, showing knock intensity, K.I., and mean indicated pressure, p_i, as functions of spark advance angle



As can be seen in the illustration, it is necessary to distinguish between three different regions on the curve of knock intensity versus spark-advance angle θ:

In region I, where $\theta < \theta_1$, a reduction of the angle θ produces a reduction in knock intensity.

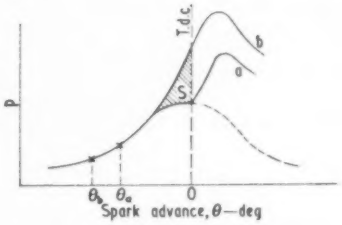
In region II, where $\theta_2 > \theta > \theta_1$, the knock intensity remains approximately constant, independent of the angle θ.

In region III, where $\theta > \theta_2$, the effect of a change in the angle θ is opposite to that obtained in region I, in that an increase in the angle θ will now produce a reduction in knock intensity.

Thus, the influence of the spark-advance angle on the knock intensity depends on the range of the angle θ. A given change in the angle may increase, or may decrease, or may not affect the knock intensity at all.

This phenomenon can be explained in the following way: When the spark-advance angle is small, as in region I of the curve, most of the mixture is burnt while the piston is already past the dead-centre position, that is to say, where the volume is large. This, of course, produces a lower maximum pressure p_z and a lower maximum temperature T_z of the gas in the cylinder. The amount of heat that is transferred during the combustion period from the gas to the large surface of the cylinder wall is also larger. For these reasons, the knock intensity falls with a decrease of the spark-advance angle.

Fig. 2. Indicator diagrams with and without negative areas above the compression curve



When the spark-advance angle is large, as in region III of the curve, most of the mixture is burnt while the piston is still before the dead-centre position. Again, the gas volume and cylinder wall surface are large, heat conduction from the gas through the cylinder wall is increased, and the maximum gas pressure and temperature are reduced. Thus, knock intensity falls with an increase of the spark-advance angle θ in the region where $\theta > \theta_2$.

When the spark-advance angle is in the intermediate region II, changes in θ tend to have no influence on the knock intensity.

The factors that determine the shape of the knock intensity curve also determine the shape of the power curve. This curve can therefore be divided, in an analogous fashion, into three separate regions. Thus, in Fig. 1, the curve showing the mean indicated pressure p_i is divided into three

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regions, I′, II′, and III′. In the region I′, where $\theta < \theta_1'$, a decrease in θ reduces p_i. In the region II′, where $\theta_1' < \theta < \theta_2'$, changes in θ have little influence on the magnitude of p_i. Finally, in the region III′, where $\theta > \theta_2'$, a decrease in θ increases p_i. The regions I′, II′, and III′ do not coincide with the regions I, II, and III, but show a relative displacement towards the left. Thus, $\theta_1 > \theta_1'$ and $\theta_2 > \theta_2'$.

This phenomenon can be explained as follows: consider the effect which, because of partial combustion taking place before the top-dead-centre position is reached, the spark-advance angle has on the negative portion of the indicator diagram. If it is assumed that, for a given spark-advance angle θ_a', the negative area above the compression curve is zero, curve *a* in Fig. 2, then it can be seen that for a larger angle θ_b a negative area S is obtained above the compression curve, as shown by the curve *b* in Fig. 2. This negative area, of course, reduces the magnitude of p_i, and thus represents negative work done in the engine. If this negative area S did not exist, the hypothetical curve p_i', dotted line in Fig. 1, would be obtained, and the regions I′, II′, and III′ would then coincide with their corresponding regions I, II, and III of Fig. 1, so that θ_2' would equal θ_2 and θ_1' would equal θ_1.

On the basis of the above facts, it is also possible to explain the phenomenon illustrated qualitatively in Fig. 3. The curve *a* shows the mean indicated pressure p_i as a function of θ, for a constant compression ratio ϵ: this is called the *iso-compression-ratio curve*. In this illustration, the curve *b* shows p_i as a function of θ, for a constant knock intensity; that is, for every value of θ, there is a different compression

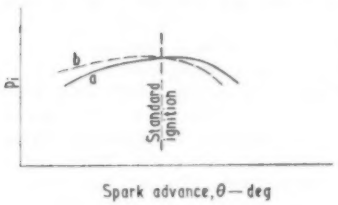


Fig. 3. Mean indicated pressure as a function of the spark advance angle: **a** *Iso-compression-ratio,* **b** *Iso-knock-intensity*

ratio ϵ: this latter curve is called the *iso-knock-intensity curve*.

The curve *a* corresponds to a compression ratio ϵ, which, for the standard spark advance angle θ_{ST}, gives a knock intensity corresponding to that of the curve *b*; this knock intensity can be at the standard value, or below it. Comparison of curves *a* and *b* in Fig. 3 shows that:

(a) When the spark-advance angle θ is smaller than the standard angle, $\theta < \theta_{ST}$, then $(p_i)_a < (p_i)_b$.

(b) For $\theta > \theta_{ST}$, we have $(p_i)_a > (p_i)_b$.

This phenomenon is explained in the same manner as that previously mentioned: for small values of θ, most of the combustion occurs after the top-dead-centre position. An increase in the compression ratio produces a faster combustion rate, so that the combustion is completed while the gas is still in a smaller volume; for this reason, losses are relatively low, so that the positive area of the indicator diagram, which, of course, represents work produced by the engine, is greater than it is for case (a). Therefore, the magnitude of p_i in the left-hand portion of Fig. 3 is greater for the curve *b* than it is for the curve *a*.

For large values of θ, however, a considerable proportion of the combustion takes place before the top-dead-centre position. Hence, an increase in the compression ratio ϵ speeds up the combustion rate, and thus increases even further that proportion of the mixture that is burnt before the top-dead-centre position is reached. This means that the negative area on the indicator diagram is increased, and the pressure p_i is decreased. In addition, since the combustion takes place earlier, the volume is greater, and so are the losses. Thus, the magnitude of p_i in the right-hand portion of Fig. 3 is greater for the curve *a* than it is for the curve *b*.

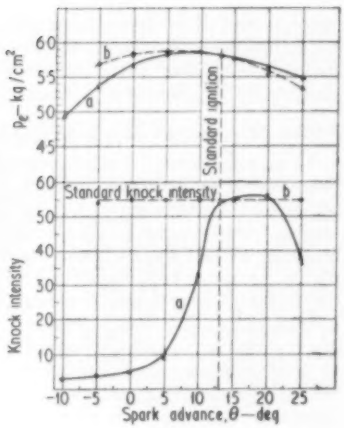


Fig. 4. Experimentally obtained curve of knock intensity and mean effective pressure as functions of the spark advance angle: **a** *Iso-compression-ratio curve,* **b** *Iso-knock-intensity curve*

Fig. 4 shows experimental results obtained by the author from a CFR Waukesha engine, for different values of the spark-advance angle. For this work the fuel used was Ethyl cracking petrol, with a nominal octane rating of 82, as determined according to the Research Method[1]. The barometric pressure was approximately 754 mm Hg. Knock intensity and mean effective pressure p_e are plotted as a function of the spark-advance angle θ.

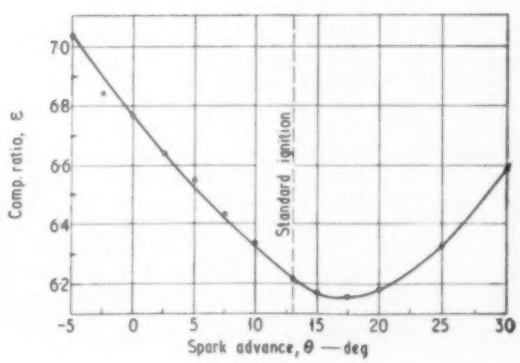
As in Fig. 3, the curves *a* are iso-compression-ratio curves; they were obtained by operating the engine at a constant compression ratio of $\epsilon = 6{\cdot}21$. The knock intensity was at the standard value of 55 units of the knockmeter, for a standard spark-advance angle of $\theta_{ST} = 13$ deg.

Again, the curves *b* are iso-knock-intensity curves, with the knock intensity maintained constant at the standard value. For each point on curve *b*, the standard knock intensity is obtained by appropriate changes in the compression ratio ϵ. Other experimental conditions of the engine were regulated in accordance with those specified by the Research Method.

In Fig. 5, the compression ratio ϵ is shown as a function of the spark-advance angle θ, with the knock intensity maintained constant at the standard value. For each spark-advance angle θ, the standard knock intensity is obtained by appropriate changes in the compression ratio ϵ. The curve shows that, without changing the knock intensity, it is possible to increase the compression ratio ϵ either by increasing or by decreasing θ.

On the basis of the experiments described, it is to be expected that, if the combustion rate could be increased by some other method, an increase or decrease of the knock

Fig. 5. Curve of compression ratio as a function of spark advance angle, obtained experimentally, with a constant knock intensity



intensity and of the power, depending on the spark-advance angle, could also be obtained. These experiments therefore provide a starting point for investigating the effect of precombustion chambers on engine operation.

Effect of precombustion chamber on knock intensity

As is well known, if the spark-advance angle is reduced below the standard value, we obtain not only a decrease in the knock intensity, but also other undesirable effects in the engine operation, such as a reduction in power, excessive heating of the exhaust valves and of the sparking plug electrodes, etc. For this reason, for normal engine operation, the spark-advance angle is not generally reduced below a certain value. To this, it must be added that the spark-advance angle is normally adjusted automatically according to the changing operating conditions of the engine. It follows that, if the angle is reduced to avoid knocking, this reduction will remain in force for all operating conditions, including even those where there is no need to reduce the knock intensity in this way.

On the other hand, if the spark-advance angle is increased to reduce the knock intensity, region III of Fig. 1, a reduction in engine power and efficiency follows and is much greater than that produced by the previous method, region I of Fig. 1, which is the one usually adopted for normal engine operation.

If the knock-intensity curve could be shifted in the direction of smaller spark-advance angles, that is, to obtain curve *m* instead of the normal curve *a* in Fig. 6, the operating conditions of the engine would be more favourable. It would then be possible to increase the spark-advance angle to obtain a smaller knock intensity, while, at the same time, the undesirable effects previously mentioned would also be reduced.

By means of a precombustion chamber, it is possible to obtain such a shifting of the knock-intensity curve in the direction of smaller spark-advance angles. In addition, the precombustion chamber reduces the height of the knock-intensity curve, for a given compression ratio. These effects are shown qualitatively in Fig. 7. Curve *a* represents ordinary engine operation, while curve *m* represents operation with a precombustion chamber. The compression ratio ϵ is kept constant in both cases.

If, with a precombustion chamber, the compression ratio ϵ is increased, then the curves m_1, m_2, m_3, etc., shown in Fig. 8, are obtained. As before, curve *a* represents normal operation, that is, without a precombustion chamber.

However, for any given spark-advance angle θ, if the compression ratio ϵ is changed until the standard knock intensity is obtained, then the curves of ϵ as a function of θ will appear as shown in Fig. 9. As before, the curves *a* and *m* represent operation without and with a precombustion chamber respectively.

As shown in Figs. 7, 8, and 9, the use of a precombustion chamber permits a reduction of the knock intensity of spark-ignition engines using a given fuel. In addition, the precombustion chamber enables the spark-advance angle to be increased, instead of decreased, for the purpose of reducing the knock intensity. Experimental results are given in subsequent illustrations and in the Tables.

Construction of the precombustion chamber

In the experiments discussed, precombustion chambers of simple construction were used, but there were no other modifications to the engine. The construction is shown schematically in Fig. 10, from which it can be seen that the chamber *A* has an external screw thread *M*, which fits into the cylinder head *G*. This screw thread can be of the same size as the thread of the normal sparking plug, for example M-18, or it could be of a different size. The sparking plug is inserted into the chamber by screwing it into the thread *N*.

A number of holes *S* are drilled through the lower part of the precombustion chamber. During engine operation, gases pass through these holes from the cylinder to the chamber, and from the chamber to the cylinder. The number of these holes can be one, two, three, or even more. Their shape, dimensions, geometrical distribution over the chamber, and the orientation of their axes, can all vary widely. Also the

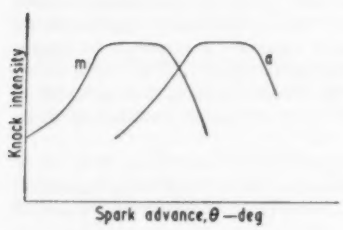


Fig. 6. The curve **m** *illustrates the desirable shift of the curve of knock intensity as a function of spark advance angle, relative to the normal operational curve* **a**

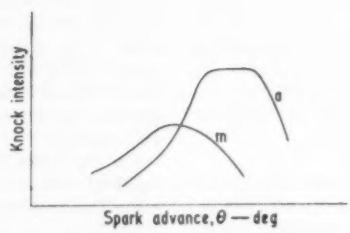


Fig. 7. Iso-compression-ratio curves showing knock intensity as a function of spark advance angle: **a** *for normal operation, and* **m** *with a precombustion chamber*

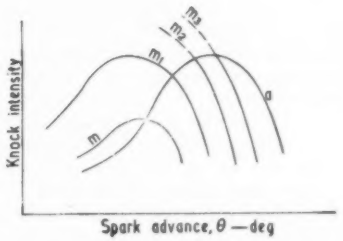


Fig. 8. Knock intensity as a function of spark advance angle: **a** *for regular operation, and* **m** *with a precombustion chamber and several different compression ratios*

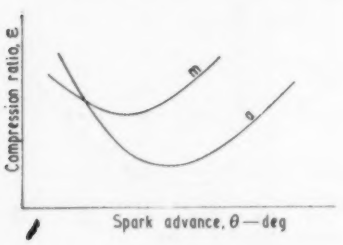


Fig. 9. Curves of compression ratio as a function of spark advance angle: **a** *for normal operation, and* **m** *when the engine is fitted with a precombustion chamber*

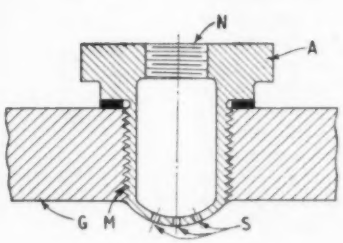


Fig. 10. Diagram showing the general arrangement of the precombustion chamber employed in the experimental investigations that are described in this article

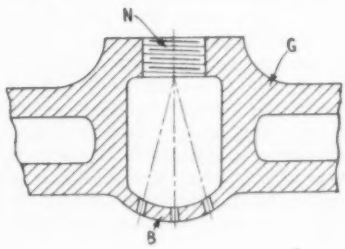


Fig. 11. Diagram of a precombustion chamber formed integrally in a cylinder head: **B** *wall of chamber, pierced by holes,* **G** *cylinder head;* **N** *sparking plug thread*

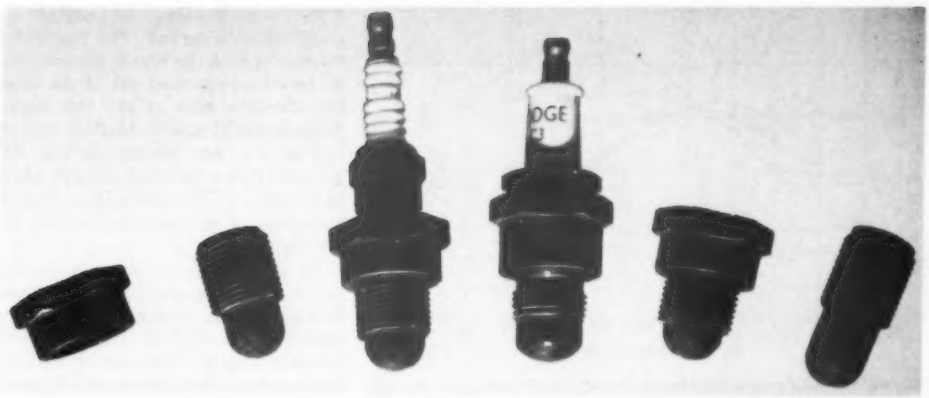
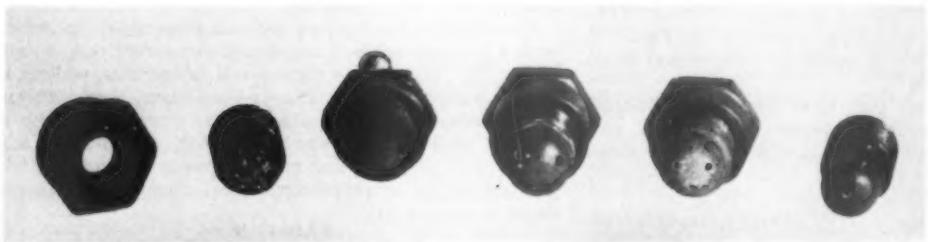


Fig. 12. A selection of the precombustion chamber designs used for the investigations



shape and volume of the chamber, and its position on the cylinder, can vary. The chamber can be made of steel, aluminium, or some other suitable metal. According to the requirements for each application, the precombustion chamber can be built with or without additional cooling or provisions for reducing the heat transfer from the chamber to the cylinder head. In construction, the chamber can be an assembly comprising several separate parts, or it can be in one piece. It can also be built as an integral part of the cylinder head, as shown in Fig. 11.

In Fig. 12 several of the chambers used in the course of the experiments are illustrated. These chambers were inserted into the cylinder head to take the place of the normal M-18 sparking plug. They each comprised several separate components. The sparking plug inserted into the chamber had an M-10 thread. No attempt was made to reduce the heat conduction from these steel chambers to the cylinder, or to provide extra cooling.

Experimental procedure

The experiments were conducted on a Waukesha CFR engine, under the conditions of the Research Method[1,2]. Ethyl cracking petrol, of the type usually used for automobiles, and with an octane rating of 82, was employed as the fuel.

This experimental work was divided into two parts. In the first, the engine was operated at its standard spark-advance angle, of 13 deg. For each run, the compression ratios were adjusted to give a standard knock intensity of 55 units, as measured on the knockmeter. The compression ratios required to give this intensity were measured for every engine condition, with precombustion chambers of various designs, and also without a chamber. In addition, the engine power was measured. Fifteen different precombustion chambers were tested in this manner. Some of these were tested in different positions obtained by turning the chamber around its axis, in order to determine the influence of the position of the holes on the gas flow inside the chamber.

In the second part of the experiment, the performance of the engine with precombustion chambers was tested for different spark-advance angles. The following were measured:

a. Compression ratio ϵ as a function of the spark-advance angle θ, with the knock intensity kept constant at the standard value.

b. Knock intensity, expressed in knockmeter units, as a function of the spark-advance angle θ, with the compression ratios ϵ kept constant. The values of ϵ used for each one of the different chambers and for operation without a chamber were obtained as follows: for every case, the value of ϵ was chosen to give the standard knock intensity when the spark-advance angle θ was adjusted to give maximum knock intensity. In addition, the knock intensity was measured as a function of θ, using the value of ϵ that corresponded to operation without a chamber.

c. Engine power, as a function of θ. This was measured both for constant knock intensity, and for constant compression ratios.

d. Indicator diagrams were also taken, so that the characteristics of the combustion curves obtained with and without precombustion chambers could be compared.

Results of the experiments

Table 1 shows the values of the compression ratio ϵ with the engine working with different types of precombustion chambers and with standard knock intensity, with a spark-advance angle $\theta=13$ deg. This value of θ is the standard ignition setting for engine operation without a precombustion chamber and under the conditions specified for the Research Method. The mixture used in these experiments was rich also as specified for the Research Method, that is, the fuel-air ratio was adjusted for maximum knock intensity. Under these conditions, it was found possible to increase the compression ratio from $\epsilon=6{\cdot}21$, without precombustion chamber, to $\epsilon=7{\cdot}11$, with chamber No. 12b. As ϵ was increased, the power and efficiency decreased, and in order

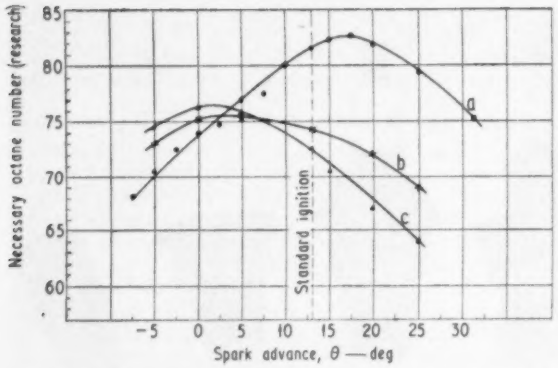


Fig. 14. Octane numbers required, plotted as a function of spark advance angle: **a** *for normal operation,* **b** *and* **c** *with precombustion chambers*

to compare the performance for operation with a chamber with that for operation without a chamber, the values of the mean effective pressure p- are also shown in the Table. A discussion on the subject of optimum compression ratios has been given elsewhere.[3]

In Table 2 are shown the values obtained for engine operation under the conditions already described, but with the precombustion chambers that gave the most significant results. The Table also shows the values of the necessary octane ratings for a constant value of 6·21 for ϵ.

To demonstrate the possibility of increasing the compression ratio ϵ and at the same time changing θ, when precombustion chambers are used, the following experimental curves were obtained, Fig. 13:

a without precombustion chamber.

b with chamber 1a.

c with chamber 9.

The knock intensity was kept constant, at the standard value. From these curves, corresponding curves can be derived to indicate the octane ratings necessary, and these are shown in Fig. 14.

As can be seen from the curves, good results can be obtained with engine operation at large values of θ. For instance, for $\theta = 21$ deg., precombustion chamber No. 9 reduced the necessary octane rating by 15 units. Accordingly, the compression ratio ϵ could be increased from 6·2 to 7·57 if the octane rating is not reduced. For $\theta = 5$ deg, the necessary octane rating and ϵ remained approximately at the same values as those corresponding to engine operation without precombustion chamber. In the experimental work the influence of θ on the knock intensity was greater than it would be in actual practical use of the engine. This is because of the characteristics of the test engine used and as well as because the Research Method test conditions were adopted.

Fig. 13 also shows curves of the mean effective pressure p_e as a function of spark advance angle θ, for the cases *a*, *b*, and *c*. The curves show that, for large values of θ, the reduction of p_e was considerable: it was in fact more than 10 deg.

In the course of the experiments, slight reductions of p_e, 1 to 3 per cent, were obtained with precombustion chambers that permitted an increase in the compression ratio ϵ from 6·21 to 6·61 and 6·68, for standard spark-advance angles. This can be seen from the Tables. As has been already mentioned, the engine was operated on a rich mixture, according to the Research Method. In other experiments, where lean mixtures were used, an increase in engine efficiency was obtained, but this phase of the problem must be dealt with separately.

Curves of the knock intensity obtained for engine operation with a constant engine compression ratio ϵ are reproduced in Fig. 15. The curve *a* corresponds to operation without a precombustion chamber. Curves *b* and *c* were obtained with chamber 1a. For curves *a* and *c*, the compression ratio ϵ was 6·2, while, for the curve *b*, it was 6·6. Fig. 16 shows the values of p_e corresponding to the curves *a* and *c* of Fig. 16. The curves for other precombustion chambers in general exhibit the same characteristics. It must be stressed that, for the experiments discussed, a significant reduction of the knock intensity was in many instances obtained with only a slight reduction of power, Table 2.

TABLE I

Experimental results for precombustion chambers

Precombustion Chamber	ϵ	p_e, $\frac{kg}{cm^2}$	$\Delta\epsilon$	Δp_e
No chamber	6·21	5·70	—	—
1b	6·70	5·48	0·49	—0·22
2	6·45	5·52	0·24	—0·18
3	6·43	5·52	0·22	—0·18
4	6·29	5·68	0·08	—0·02
5	6·62	5·39	0·41	—0·31
6	6·59	5·41	0·38	—0·29
7	6·68	5·52	0·47	—0·18
8	6·34	4·93	0·13	—0·77
9	6·79	5·33	0·58	—0·37
10	6·49	5·35	0·28	—0·35
11	6·70	5·40	0·49	—0·30
12b	7·11	5·00	0·90	—0·70
13	6·43	5·25	0·22	—0·45
14	6·61	5·63	0·40	—0·07

The experimental conditions were: Standard knock intensity of 55, Research Method, for standard spark-advance angle of 13 deg and standard fuel-air ratio of 1·5 in. The fuel was Ethyl cracking petrol with nominal octane number, RON, of 82.

TABLE II

Experimental results for certain precombustion chambers

Precombustion chamber	*Compression ratio ϵ, for RON=82·4*	*Required octane No., RON, for ϵ=6·21*	*Reduction of required octane No., Δ RON*	*Power N_e per cent*
No chamber	6·21	82·4	—	100
14	6·61	75·9	6·5	99
7	6·68	74·9	7·5	97
1b	6·70	74·8	7·6	96
11	6·75	74·1	8·3	95
5	6·62	75·7	6·7	95
9	6·79	73·7	8·7	93
12b	7·11	70·6	11·8	88

Standard knock intensity of 55, Research Method. Fuel as for Table I.

These results can be explained in the following manner: The precombustion chamber induces additional mixing of the petrol and air, thus causing on one hand, an increase in the combustion rate and, on the other, an increase in the rate of heat conduction from the gases to the wall of the cylinder head. These two phenomena reduce the knock intensity in the range of θ used in practice for engine operation, that is, in the range for which most of the combustion takes place while the piston is not far from the top-dead-centre position.

If most of the combustion takes place while the piston is far from top-dead-centre, an increase in the combustion rate will produce the phenomena discussed towards the beginning of this paper. For this reason, the use of a precombustion chamber might not reduce the knock intensity for small values of θ.

The reduction of the power produced by the precombustion chamber can be explained as follows. When a precombustion chamber is used, the heat losses are greater because of the increase in conduction to the cylinder head. As is known,

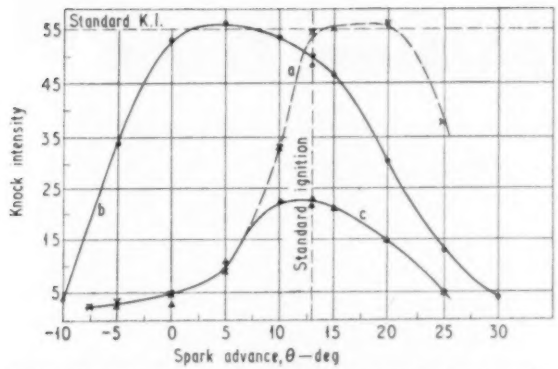


*Fig. 15. Knock intensity plotted as a function of spark advance angle: **a** normal operation and $\epsilon = 6{\cdot}2 : 1$; **b** with precombustion chamber and $\epsilon = 6{\cdot}6 : 1$; **c** with the same precombustion chamber and $\epsilon = 6{\cdot}2 : 1$*

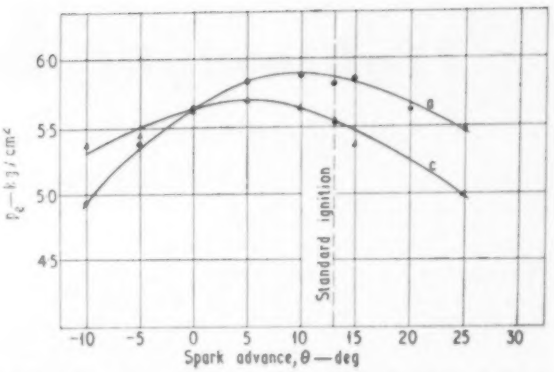


*Fig. 16. Iso-compression-ratio curves showing mean effective pressure as a function of spark advance angle: **a** normal operation; **c** with a precombustion chamber installed in the cylinder of the engine*

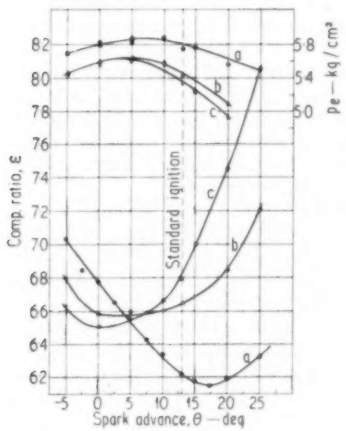


*Fig. 13. Curves, determined experimentally, of compression ratio and mean effective pressure as functions of spark advance angle: **a** for normal operation, **b** and **c** for operation with the precombustion chambers*

such heat losses are also obtained with diesel engines using precombustion chambers.

Recently, experiments on spark-ignition engines with precombustion chambers have also been made by Nilov[4], to obtain a reduction in fuel consumption. In these experiments, two different types of chambers were used. With the first, an additional carburetter and an additional valve, operated by a cam-shaft, were employed. The second was used with a piston type compressor. Nilov obtained a reduction in power and efficiency for rich, stoichiometric mixtures, and an increase in efficiency for lean mixtures, which is in agreement with the results of the experiments described here. It must be stressed again that the precombustion chambers used by the author are of simple construction, and require no engine modifications or additional apparatus, and yet they give a considerable reduction of the knock intensity.

General conclusions

On the basis of the experiments, it can be stated that the use of precombustion chambers on spark-ignition engines gives a considerable reduction of the knock intensity. These results depend on:

a. Proper construction of the chamber for the type of engine used, and

b. Amount of the spark-advance angle θ.

It was found that, within the range of θ generally used in engine operation, reduction of the knock intensity by means of a precombustion chamber is more significant for large than for small angles. For angles of θ smaller than those generally used, the precombustion chamber is liable to produce insignificant results, and may even cause an increase in the knock intensity. Whenever a precombustion chamber is used, the spark-advance must be different from that for regular engine operation. If necessary, the knocking can be eliminated by increasing the angle θ, instead of by decreasing it.

In the experiments described, a decrease in power and efficiency, at rich mixtures, took place when a precombustion chamber was used. However, for many of the precombustion chambers, this decrease was small, 1 to 5 per cent. It is to be expected that, if more refined designs of chamber are used, this decrease of power and efficiency will be obviated, or that the power and efficiency may even increase. But to attain this, more experiments with precombustion chambers must be carried out, using different types of engines. It is also necessary to determine, for every type of engine, the optimum spark-advance angle for any given type of precombustion chamber.

The author wishes to acknowledge the aid of Mr. J. Gildor and Mr. W. Karfunkel, who helped in the experimental phase of the work.

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2. ROSSENBECK: "Die Oktanzahl als Wertmasstab neuzeitlicher Ottokraftstoffe", *A.T.Z.* No. 9, 1958.
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Stud Welding

A NEW power source for stud welding has been introduced by Crompton Parkinson (Stud Welding) Ltd. Designated the TRS3, the new unit is designed to provide all types of Crompton Parkinson stud welding equipment with the D.C. power required for the stud welding of ferrous attachments of $\frac{3}{8}$ in diameter or less. This power source is claimed by the manufacturers to provide almost noiseless operation, good arc stability, high power factor on load and high electrical efficiency.

An alternative version of the new unit, designated the TRS3C, is available for use where stud welding of non-ferrous as well as ferrous materials is required. It is basically the same as the standard TRS3 but has an in-built C type regulator. Both these rectifiers are on wheels and of compact dimensions, with a weight of little over 400 lb.

Lightweight Amphibian Truck

Chrysler Corporation and Reynolds Metals Company Collaborate in Development of an Aluminium, 8×8, 2½-tonner for the U.S. Army

WITH its current commitments, the U.S. Army is faced with global logistical problems. Its surface transport equipment must be capable of operating over any terrain, in any part of the world, and under any climatic conditions. In view of the quantities required, large-scale production facilities must be in existence and capable of rapid expansion since, in emergency, wastage is likely to occur at a high rate. Obviously, the range and quantity of spares required to maintain vehicles in the field is of major importance as a world-wide service must be organized.

Concepts of conditions in any war in the foreseeable future differ radically from those obtaining in the past. Tactical missiles will, it is thought, extend the depth of a battle line from the earlier 5 miles to about 150 miles. Consequently, while a vehicle must have a good road performance it must also be able to travel over rough country or in devastated areas. Further, it must not be held up by natural water barriers and should not be dependent on fords or bridges, which inevitably constitute bottlenecks and are targets for enemy fire or attack.

Always an important consideration in battle, time is likely in the future to be a critical factor. It follows that, to be effective, men and materials will have to be transported long distances or over devastated or contaminated areas in the shortest possible time. Therefore, the vehicles and equipment, like the men, must be transported by air and, if necessary, parachuted from the air.

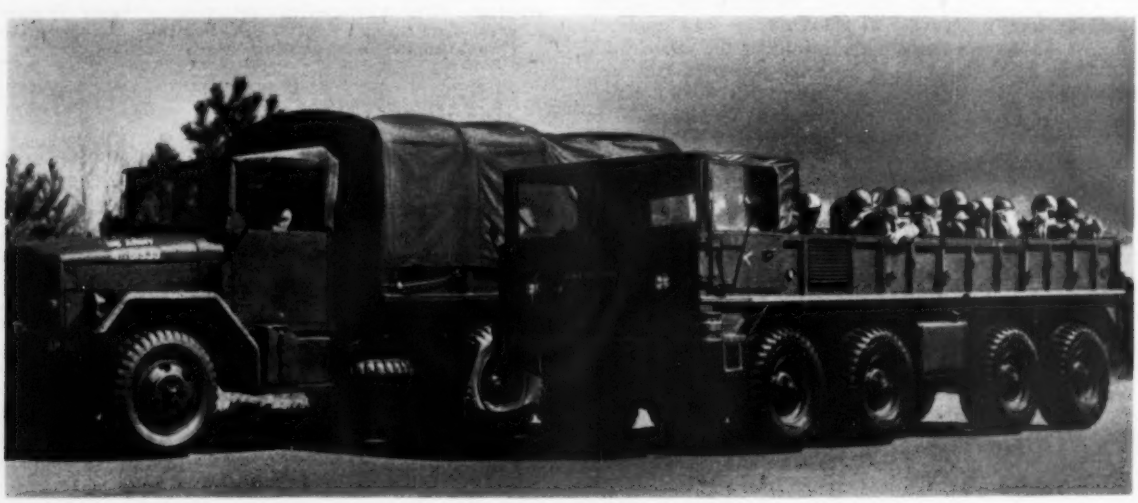
Against the background of these, and other, formidable requirements, the U.S. Army authorities framed a specification and awarded a contract to the Chrysler Corporation to develop a 2½-ton truck to replace the current standard Army vehicle of that capacity. To the credit of all concerned, a truly remarkable vehicle designed to meet the specification has been tested and, earlier this year, demonstrated under simulated operational conditions before the competent military authorities. Fully detailed particulars of the vehicles have not yet been released but the information at present available merits consideration.

In the early stages of project design a number of major decisions were made. (1) To ensure minimum gross weight, the fullest use be made of aluminium in construction. (2) While basically a land vehicle, the truck to be amphibious. (3) Although wheeled for good road performance, the unit ground pressure not to exceed that of a comparable tracked vehicle. (4) For cross-country performance, fully independent suspension, all-wheel drive, and power steering be incorporated. (5) To save initial expenditure and time and to minimize hazards in the future production of vehicles and spares, the widest use be made of standard commercially-produced mechanical units such as engines, transmissions, drive shafts, axles, and other components.

Participation of Reynolds Metals Co. in the project dates from November 1957. Dimensions and loadings stipulated by Chrysler project engineers were received by specialists of that company, who then provided design suggestions and metal specifications for a "unitized" body and chassis and for other component parts. Interlocking extrusions were adopted for the flooring of the body, and extruded sections were also used for longitudinal members, side-rails, and cross members. Sheet metal was used for side panelling and for various other parts. Both welding and riveting were employed in construction. The test model illustrated embodies about 1,700 lb of Reynolds aluminium.

Floatability with a full-capacity load for amphibious operations is provided by the hull-type construction of the unitized body and chassis. With suitable auxiliary equipment—winch or marine propeller—the vehicle can be winched or propelled across a river under its own power.

The new lightweight truck, shown ranged alongside a standard U.S. Army 2½-tonner, weighs less fully loaded than the current vehicle unladen



Independent torsion bar suspension on each of the eight driven wheels gives the truck an excellent cross-country performance. A power-assisted steering system actuates the front four wheels. These views show the vehicle climbing up from the strand after crossing a lake

Alternatively, it can be propelled by a standard, commercially built, outboard motor mounted on the tailgate; towed by a powered boat; or towed by a helicopter. Incidentally, this hulled construction is expected to lead to a substantial reduction in maintenance costs. Engine, transmission units, drive shafts, brakes, and axles are all contained within the hull and are thus protected against grit, mud, and water. Such protection will extend the periods between necessary servicing and prolong the life of the components.

The saving in weight effected by the use of aluminium and the unitized construction of body and chassis makes it possible for the truck to carry a payload in higher proportion to its weight than is possible with more conventional vehicles. On cross-country operations it can carry a load approximately two-thirds of its weight as compared to one-third to one-half in the case of vehicles in current use. Probably, the most striking feature of this aspect of weight saving is that the new truck fully laden weighs less than the currently standard U.S. Army truck, of similar capacity, in the unladen condition.

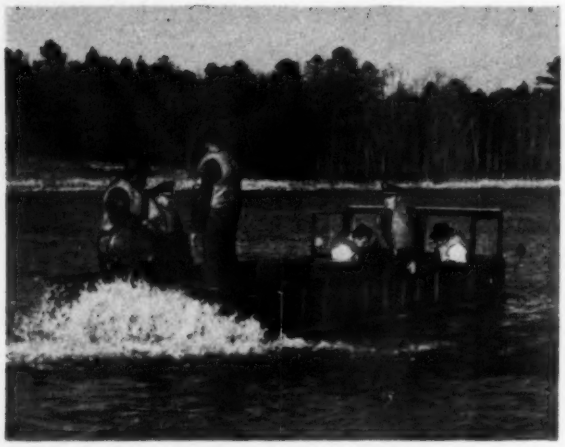
The decision to design the vehicle so as to make use of existing facilities for the production of the standard Chrysler 361 in^3 (5,916 cm^3) capacity, 8-cylinder engines, transmission units, drive shafts, and axles, will reduce by many millions of dollars the cost that otherwise would confront the U.S. Government for such facilities and the necessary tooling. In general, the manufacture of the new vehicle can readily be adapted to commercial techniques and processes.

Designed for operation over virtually any terrain, the truck has eight wheels, all driven, with independent suspension by torsion bars all round. Steering on the front four wheels is power-assisted for ease of control. This gives it the cross-country versatility of a tank or a fully tracked carrier. The wheels are shod with wide-section, low-pressure tyres and the specific ground pressure is lower than that of any tank and of many track-laying vehicles. Nevertheless, maintenance costs are substantially lower and the truck can also operate at normal highway speeds; an advantage not possible with any track-laying vehicle.

Unladen, the truck weighs 8,600 lb, approximately half the weight of the U.S. Army's present standard 2½-tonner. In overall length it is about 2 ft 6 in shorter and in height somewhat lower. Its lightness and compactness make it readily transportable in existing military cargo aircraft. Thus, it can be flown rapidly to distant strategic areas and, if necessary, parachuted on reaching its destination.

Functionally, it will be a multi-purpose vehicle. With

Shown here equipped as a troop carrier, one of a number of versions, the new aluminium amphibian truck was demonstrated recently before United States Army officials



The fully laden vehicle can float. It may be propelled under its own power or by an outboard motor, or towed by a boat or a helicopter

suitable modification of the body configuration it will serve as a troop carrier, cargo carrier, fuel or water tanker, missile launcher, ambulance, or mobile crane. The unusually high order of interchangeability of parts and components will make possible a substantial reduction in the number of spare parts required for servicing. It is estimated that the number of spare part items currently required on a world-wide scale for servicing comparable vehicles can be reduced by 33 per cent.

As regards the impact made by the new truck on the U.S. Army authorities, it needs only to be reported that the Chrysler Corporation is currently engaged in the development of a 6×6, 1½-tonner and a 4×4, 1-tonner. These will be on precisely similar lines and incorporate, as far as is practicable, the same components in order to complete a "family" of vehicles for military purposes.

In what may fairly be described as a revolutionary change-over by the U.S. armed forces to lightweight, airborne, low-maintenance equipment to meet future requirements, aluminium figures to an unprecedented extent. Aluminium armour plate can give good protection but at considerably less weight than steel armour. Further, aluminium has a "shorter half-life". In other words, it retains radiological contamination for a shorter period of time than many other materials. Several military vehicles of outstanding interest have been developed.

An instance of how weight-saving by the use of aluminium is turned to great effect in armoured vehicles is shown by the M-113 aluminium-armoured personnel carrier; a fully tracked, enclosed vehicle. Compared with the M-59, its steel-armoured predecessor, weight is reduced from 21 to 10 tons, range is extended from 120 to 200 miles, speed is increased from 32 to 40 m.p.h., and fuel consumption rate from 1·0 to 2·6 mile/gal.

The greatly reduced weight of the vehicle has enabled it to be powered by one instead of two engines, with a resulting saving in maintenance. Overall it is shorter, narrower and lower, yet can carry the same infantry squad of twelve combatants plus the driver and the commander. It can be carried in a cargo aircraft, can be parachuted to a battlefield area, and can negotiate inland waterways. The aluminium armour will give some protection against the heat and blast of atomic explosions as well as of conventional ones.

New British Cars

SUMMARY OF THE NOTEWORTHY FEATURES OF RECENTLY INTRODUCED VEHICLES

Austin Healey 3000

DURING recent years, the handsome Austin Healey sports car has established itself firmly by virtue of its alliance of high performance with tractability and good behaviour on the road. These qualities should be enhanced on the latest version, known as the Austin Healey 3000. As the name suggests, a major change in the specification is the installation of a 3-litre engine in place of the 2·6-litre unit employed on the superseded model 100-Six. To match the increased road potential with greater stopping power, Girling disc brakes replace drum brakes on the front wheels. Two basically similar open bodies will be available, one a two-seater and the other an occasional four-seater.

As before, the six-cylinder engine is based on the familiar B.M.C. Series C unit. While the stroke of this remains at 88·9 mm, however, the bore has been increased from the previous 79·4 mm to 83·34 mm, giving a swept volume of 2,912 cm³ as against 2,639 cm³; as a result of this change, the stroke : bore ratio has been reduced from 1·12 : 1 to 1·07 : 1. The standard compression ratio has been raised from 8·5 : 1 to 9·03 : 1. There has been no alteration to the four-bearing crankshaft, nor to the automatically tensioned chain drive to the camshaft, which actuates the vertical overhead valves through a conventional arrangement of push rods and rockers. Also virtually unaltered from that of the 100-Six, the cylinder head differs from that of the

In appearance, the Austin Healey 3000 closely resembles its predecessor

standard Series C engine in not having an integral inlet manifold. Instead, a cast aluminium manifold, feeding individual inlet ports, is bolted to the head. This manifold carries two S.U. semi-downdraught HD6 carburettors, which are fitted with pancake type air filters. The carburettors are shielded by a plate from direct heat radiation from the two three-branch exhaust manifolds, each of which has its own outlet pipe and silencer.

A modified bath tub form of combustion chamber, embodying Weslake patents, is employed. Because the main

object of the increase in swept volume was to improve the torque rather than the power output, no change has been made to the sizes of the valves and ports. The new engine has a maximum gross power output of 130 b.h.p. at the moderate speed of 4,750 r.p.m., equivalent to 44·7 b.h.p/litre. Its maximum gross torque is 175 lb-ft at 3,000 r.p.m., so the b.m.e.p. is 149 lb/in². For the 2·6-litre Austin Healey engine, the respective figures were 117 b.h.p., 149 lb-ft and 140 lb/in², at the same speeds. It is therefore noteworthy that, although the swept volume has increased by just over 10 per cent, and has resulted in an 11 per cent rise in power output, the improvement in maximum torque is over 17 per cent. This reflects the rise of about 6½ per cent in the b.m.e.p. figure. According to the makers, this increase is not the result of any special development work, but is attributable to a more satisfactory ratio of piston area : valve opening area.

The ratio of connecting rod length : stroke is 1·89 : 1, and the mean piston speed at maximum b.h.p. is 2,770 ft/min. At maximum torque, the speed is 0·63 of that at maximum power. A relatively flat torque curve is indicated by the ratio of 1·185 : 1 between the maximum torque and the torque at maximum power. The minimum brake specific fuel consumption is satisfactorily low at 0·505 pt/b.h.p-hr. Other performance criteria of interest are the figures of 2·56 b.h.p/in² piston area, 0·214 b.h.p/lb engine dry weight and 116·5 b.h.p/ton vehicle kerb weight.

In the Borg and Beck clutch, a friction area of 2×39 in² is afforded by the 10 in diameter plate. To withstand the higher torque output of the new engine, the four-speed gear cluster has been strengthened by an increase in diametral pitch. Also, first, third and reverse ratios have been raised and second has been lowered. The actual ratios are 1 : 1, 1·309 : 1, 2·053 : 1 and 2·93 : 1, with 3·78 : 1 reverse, whereas those previously used were 1 : 1, 1·333 : 1, 1·913 : 1 and 3·076 : 1, with a reverse of 4·176 : 1. The higher third gear will probably be beneficial on the road, and the additional engine torque will more than cope with the alteration to bottom gear. Although second gear now appears to be undesirably low, in that it is closer to bottom gear than to third, it is often difficult to obtain the optimum spacing of the intermediate ratios without recourse to a non standard diametral pitch. There is synchromesh on second, third and top gears, and the gear change is controlled by a short, central lever of remote control type.

Laycock-de Normanville overdrive can be supplied as an optional extra and, when it is fitted, a final drive ratio of 3·91 : 1 is employed instead of the standard ratio of 3·545 : 1. The overdrive step-up is 0·822 : 1 and it is effective on third gear as well as top, giving an overdrive third ratio of 1·076 : 1. This is so close to direct top gear that its only merit would appear to be its virtually instantaneous engagement, by means of the switch, from normal third—useful during high-speed overtaking. The propeller shaft is of the conventional open type, and the hypoid-bevel rear axle has three-quarter floating half-shafts. With the standard axle ratio, the speed in top gear per 1,000 engine r.p.m. is 20·9 m.p.h. Where overdrive is fitted, the corresponding speeds are 18·94 m.p.h. in direct drive and 23·1 m.p.h. in overdrive.

Although the unladen weight distribution of the two-seater is almost balanced at 52 per cent on the front wheels and 48 per cent on the rear wheels, the occupants are seated to the rear of the car's centre of gravity and the fuel load may be up to 12 gallons. Also, as will be described later, the four-seater has an overhung stowage of the battery and spare wheel, in addition to its inevitably more rearward disposition of the centre of gravity of its full passenger load. To avoid oversteer as a result of this bias of weight distribution towards the rear wheels, a degree of roll understeer is induced by mounting the semi-elliptic rear springs lower at the front than at the rear.

The clean lines of the body are apparent in this view. Disc type front brakes have been adopted as standard, but the wire wheels are an extra. Also available at additional cost is a detachable hardtop, constructed of glass reinforced plastics

In the normally laden condition, the rear springs have a slight negative camber. The axle is located laterally by a Panhard rod, and spring movement is controlled by lever type hydraulic dampers. A spring rate of 156 lb/in is used at the rear, and the suspension periodicity with the vehicle unladen is 94 c/min. The roll centre height is 9⅜ in, and the roll stiffness is 2,020 lb-in/deg.

For the front suspension, a double transverse wishbone layout is employed, in conjunction with coil springs, lever type dampers and an anti-roll bar. The figures for spring rate, periodicity, roll centre height and roll stiffness are respectively 66·5 lb/in, 89 c/min, 4 1/16 in and 6,990 lb-in/deg. A cam-and-peg type steering gear, with a reduction of 14 : 1, is fitted; the three-piece track rod and slave lever layout conforms with normal practice. Lock to lock movement of the front wheels involves approximately three turns of the steering wheel, and the turning circle is 35 ft.

The Girling rear brake installation of the 100-Six model, with its 11 in diameter drums and 2½ in lining width, is unchanged on the new car. As would be expected, a leading and trailing shoe layout is employed. The front brakes have discs of high-grade cast iron; they are of 11¼ in diameter and ⅜ in thick. Each caliper unit incorporates a hydraulic cylinder on each side of the disc, actuating a segmental friction pad of the quickly detachable type. The total area of the two pads is 8·5 in², and a self-adjustment feature is incorporated, so that wear is compensated for automatically. Of the total braking effort applied by the master cylinder, 57·4 per cent is applied to the front wheels and 42·6 per cent to the rear wheels. Ventilated, steel disc type wheels are standard equipment, but centre-lock wire wheels are available at

extra cost; in each case, the wheels have 15 in diameter rims and are fitted with tyres of 5·90 in section.

There have been no changes of any magnitude to the partially integrated chassis and body construction. The frame is based on two full-length, box-section members, relatively close together and almost parallel in plan. Ahead of the power unit is a box-section cross member that supports the front suspension. Behind the gearbox is a cruciform member, also of box section, and at the rear of the longitudinal members is another cross member carrying the shackles of the rear springs. The front eyes of the springs are attached to outriggers which, together with further outriggers in line with the clutch housing, form supports for the body side rails. Made in two pieces, the floor is welded to the longitudinal members and the side rails, which are made into box section by the lower sides of the body.

The scuttle, wheel arches, door sills and body framing all form part of the basic unitary structure. To facilitate repairs, the wings are detachable and, like the doors, bonnet and boot lid, are of steel. A considerable saving in weight is effected by the use of aluminium for the rest of the body, namely, the front and rear sections of relatively simple shape between the wings. The bonnet lid is hinged at the rear edge and embodies a small raised air intake opening near the front, to assist ventilation of the engine compartment.

On both bodies, separate bucket type front seats are provided. The rear seats of the occasional four-seater version have separate pans and cushions but a common squab. On this model, the spare wheel is housed horizontally on the floor of the boot, and the 12-volt battery is stowed in the same compartment, behind the right-hand wheel arch. The two-seater, however, has its spare wheel mounting on the box over the rear axle, and the two 6-volt batteries are contained within this box. On the transmission tunnel, between the front seats, is a padded armrest.

The removable hood is of p.v.c.-coated fabric and embodies a large plastics rear light; the hood sticks also can be completely detached if desired. A tonneau cover, of similar material to the hood, and detachable side lights are standard equipment. The side lights have polished aluminium frames and each comprises two Perspex panels, one fixed and one sliding. For each type of body, a removable hardtop of glass reinforced polyester resin is available at extra cost. It embodies a wrap-round, three-piece rear light of moulded Perspex. Other optional items in the specification include a heater and a radio set.

In the specification data panel will be found the principal dimensions of the car. The ratio of wheelbase : mean track is 1·87 : 1 and that of wheelbase : overall length is 0·584 : 1. This ratio is greater than the appearance of the vehicle suggests, because of the appreciable projection of the bumpers and overriders beyond the body. The relatively small lateral overhang of the body is indicated by the ratio of mean track : overall width of 0·816 : 1.

Wolseley 6/99

WITH the introduction of the Wolseley 6/99, the association of Pinin Farina with the British Motor Corporation has been extended to embrace the larger type of car. The new model, a commodious 2·9-litre vehicle of modern appearance, supersedes the 6/90; its resemblance to the 1½-litre 15/60, announced in December 1958, is sufficiently marked to make it instantly recognizable as a Wolseley. Like the smaller car, the 6/99 is of unitary construction, and the basic layout follows orthodox lines. The power unit is the new B.M.C. Series C engine with an increased bore, from which was developed the Austin Healey engine described in a preceding section of this article.

A departure from tradition is the choice of a three-speed gearbox on the standard model, with synchromesh on all forward gears. Five forward ratios are provided, however, by the use of an overdrive unit, effective on top and second gears. Automatic transmission is available as an optional extra. The rapidly increasing popularity of the disc brake is reflected in its adoption for the front wheels of this vehicle.

As has already been indicated, the increase in the capacity of the six-cylinder Series C engine, from 2,637 cm³ to 2,912 cm³, has been obtained by enlarging the bore from 79·4 mm to 83·34 mm. The version installed in the Wolseley has the standard cylinder head, with bath tub type combustion chambers and an integral, water jacketed inlet manifold. Fitted to the manifold are two S.U. H4 horizontal carburettors, connected by a cast aluminium duct to the oil-bath air filter. Fuel is delivered by two S.U. PD electric pumps mounted on the rear face of the front wall of the boot. Although this arrangement has been criticized in the past, on account of the audibility of the pumps to the passengers in the rear seats, it has the advantage of virtually eliminating the risk of vapour lock, because the pumps are in a relatively cool place and are pushing, not sucking, the fuel. Moreover, if the carburettor float chambers are empty, after a long period of standing, the pumps begin

Styled by Pinin Farina, the new 2·9-litre Wolseley 6/99 bears a marked resemblance to the 15/60 model introduced late last year

to fill them immediately the ignition is switched on. The use of two pumps provides a useful safeguard: in the event of failure of one, the delivery rate of the other is sufficient for two-thirds throttle to be maintained.

Whereas the standard compression ratio is 8·3 : 1, the alternative of 7·2 : 1 is available for certain overseas countries where the fuel quality is low. With the higher ratio, the maximum gross power output is 112 b.h.p. at 4,750 r.p.m.; and with the lower ratio, the output is reduced to 109 b.h.p. These powers are equivalent to 38·5 and 37·5 b.h.p/litre. On the basis of the higher output, the b.h.p/in² piston area is 2·21, and the b.h.p/lb engine weight and b.h.p/ton vehicle kerb weight are respectively 0·187 and 73·5.

The mean piston speed and the ratios of stroke:bore and of connecting rod length : stroke are as detailed for the Austin Healey 3000. Again on the higher compression ratio, the maximum gross torque is 158 lb-ft at 2,000 r.p.m. The ratio of speed at maximum torque : speed at maximum b.h.p. is thus 0·42 : 1, and the ratio of maximum torque : torque at maximum b.h.p. is 1·28 : 1. At maximum torque, the b.m.e.p. reaches the satisfactory figure of 134 lb/in².

With the standard, manually controlled transmission, the drive is transmitted by a Borg and Beck single-plate clutch. It has a diameter of 10 in and a friction area of 2×39 in²; hydraulic actuation has been retained. The gearbox is controlled by a lever on the steering column. Entirely new synchromesh mechanisms, of the baulk ring type, are embodied. The ratios are 1 : 1, 1·65 : 1 and 3·1 : 1, with a reverse of 3·01 : 1. A push-pull handle, below the facia, actuates the hydraulic control of the Borg-Warner overdrive unit. Since this unit provides a step-up of 0·7 : 1, the overdrive ratio in second gear is 1·16 : 1, which seems to be rather close to the normal top gear ratio.

The optional automatic transmission is the familiar Borg-Warner unit, on which there are a low and an intermediate ratio, variable by means of a hydraulic torque converter, and a direct-drive top gear. A hydraulic control system changes the ratio as required by the speed and load conditions. An open propeller shaft drives the hypoid-bevel

Lockheed disc brakes have been adopted for the front wheels. The cast iron discs have a diameter of 10·8 in and are ½ in thick

three-quarter floating rear axle. The final drive ratio is 3·9 : 1 with the normal transmission and 3·545 : 1 with the automatic unit. In top gear, the road speeds per 1,000 engine r.p.m. are 20.97 m.p.h. for the automatic model, and 19·02 and 27·25 m.p.h. in normal and overdrive for the other.

An orthodox rear suspension layout of semi-elliptic springs is employed, and their movement is controlled by lever type dampers. Less usual on a car of this kind is the use of an anti-roll bar at the rear as well as at the front. This is indicative of a well balanced weight distribution and therefore basically sound conditions for good handling characteristics. The front suspension is by coil springs and double transverse wishbone links, the upper wishbones being mounted on the spindles of the lever type dampers. A spring rate of 185 lb/in is specified at the rear, and the corresponding figure for the front is 180 lb/in, measured at the wheel. The respective periodicities, unladen, are 76 and 102 c/min.

A steering gear of cam-and-peg type is employed in conjunction with a conventional three-piece track rod and slave lever assembly. The ratio of the gear is 20 : 1. Movement of the wheels from lock to lock requires four turns of

Austin Healey 3000 SPECIFICATION DATA

Engine

Number of cylinders 6
Bore 83·34 mm
Stroke 88·9 mm
Swept volume 2,912 cm³
Maximum b.h.p. (gross) 130 at 4,750 r.p.m.
Maximum b.m.e.p. 149 lb/in²
Maximum torque 175 lb-ft at 3,000 r.p.m.
Compression ratio 9·03 : 1
Crankshaft Four-bearing, forged steel
Valves Push rod operated overhead valves; KE965 exhaust valves
Combustion chamber type Modified bath tub with vertical valves
Carburettors Two S.U. HD6, semi-downdraught
Fuel pump S.U. electric
Dry weight 608 lb

Transmission

Clutch Borg and Beck single dry plate, 10 in diameter
Gearbox type Four forward speeds, with synchromesh on second, third and fourth
Gear ratios

	Normal	Overdrive
fourth	1 : 1	0·822 : 1
third	1·309 : 1	1·076 : 1
second	2·053 : 1	
first	2·93 : 1	
reverse	3·78 : 1	

Dry weight, with clutch bell housing
standard 79 lb
with overdrive 116½ lb
Propeller shaft Open, with needle roller bearing universal joints

Rear axle

Type Three-quarter floating, with hypoid-bevel reduction
Ratio standard 3·545 : 1
with overdrive 3·91 : 1

Suspension

Front Double transverse wishbones and coil springs, with lever type dampers and anti-roll bar
Rear Semi-elliptic springs, with Panhard rod and lever type dampers

Steering

Make and type Cam Gears cam-and-peg
Ratio 14 : 1
Turning circle 35 ft
Turns from lock to lock 3 approx.

Brakes

Front Girling hydraulic, disc type, 11¼ in diameter
Rear Girling hydraulic, drum type, 11 in diameter; shoe width, 2¼ in; mechanical actuation by hand brake
Distribution of braking effort
front 57·4 per cent
rear 42·6 per cent

Wheels and tyres

Wheel type Ventilated disc, steel, with five-stud attachment. Centre-lock wire type optional extra
Tyre size 5.90—15.00 in
Pressures front 20 lb/in²
rear 23 lb/in²

Dimensions

Wheelbase 7 ft 8 in
Track, front 4 ft 0⅛ in
Track, rear 4 ft 2 in
Ground clearance 4½ in
Overall length 13 ft 1½ in
Overall width 5 ft 0½ in
Overall height 4 ft 2 in with hood raised
Kerb weight, with overdrive and full fuel tank
two-seater 2,498 lb
four-seater 2,483 lb

Wolseley 6/90 SPECIFICATION DATA

Engine

Number of cylinders 6
Bore 83·34 mm
Stroke 88·9 mm
Swept volume 2,912 cm³
Maximum b.h.p. (gross) 112 at 4,750 r.p.m.
Maximum b.m.e.p. 134 lb/in²
Maximum torque 158 lb-ft at 2,000 r.p.m.
Compression ratio 8·3 : 1
Crankshaft Four-bearing, forged steel
Valves Push rod operated, overhead, vertical
Combustion chamber type Bath tub
Carburettors Two S.U. H4, horizontal
Fuel pump Two S.U. PD electric
Dry weight 598 lb

Transmission

Clutch Single dry plate, 10 in diameter, with standard transmission only
Gearbox type Three-speed, with synchromesh on all forward ratios; Borg-Warner overdrive. Borg-Warner automatic transmission optional
Gear ratios, standard

	Normal	Overdrive
third	1 : 1	0·7 : 1
second	1·65 : 1	1·16 : 1
first	3·1 : 1	
reverse	3·01 : 1	

Gear ratios, automatic
intermediate 1·435 : 1
low 2·309 : 1
reverse 2·005 : 1
torque converter range 2·15 : 1
Dry weight
standard 132 lb
Propeller shaft Open, with needle roller bearing universal joints

Rear axle

Type Three-quarter floating, with hypoid-bevel reduction
Ratio standard 3·9 : 1
automatic 3·545 : 1

Suspension

Front Double transverse wishbones and coil springs, with lever type dampers and anti-roll bar
Rear Semi-elliptic springs, with lever type dampers and anti-roll bar

Steering

Make and type Cam Gears cam-and-peg
Ratio 20 : 1
Turning circle 40 ft
Turns from lock to lock 4

Brakes

Front Lockheed hydraulic, disc type, 10·8 in diameter
Rear Lockheed hydraulic, drum type, 10 in diameter; shoe width, 3 in

Wheels and tyres

Wheel type Ventilated disc, steel, with five-stud attachment
Tyre size 7·00—14 in tubeless
Pressures, fully laden
front 28 lb/in²
rear 28 lb/in²

Dimensions

Wheelbase 9 ft
Track, front 4 ft 5⅛ in
Track, rear 4 ft 5½ in
Ground clearance 6¼ in
Overall length 15 ft 8 in
Overall width 5 ft 8½ in
Overall height 5 ft 0 in
Kerb weight, with full fuel tank
standard 3,421 lb

The boot of the 6/99 is unusually capacious and the tail fins are restrained in styling

the steering wheel and the 40 ft turning circle is probably reasonable for the size of vehicle.

As indicated earlier, drum and disc brakes are employed, the system being of Lockheed manufacture. At the rear, a leading and trailing shoe layout, in 10 in diameter drums, has been adopted, and the lining width is 3 in. Discs of 10·8 in diameter and $\frac{1}{2}$ in thickness are employed on the front wheels and, because of the weight of the car and the absence of self-servo effect with disc brakes, the pedal effort is augmented by a vacuum servo. For changing the friction pads, the calipers can be withdrawn on removal of the two mounting bolts.

The pistons of the disc brakes embody a means of self adjustment, and another interesting feature of the braking system is the use of a pressure limiting valve in the fluid line to the rear brakes. This reduces the risk of the wheels locking under heavy retardation. When the valve closes, at a predetermined pressure, any additional pedal effort is transmitted to the front brakes only. The vacuum servo unit is of the Lockheed suspended vacuum type; it has a diameter of $6\frac{1}{4}$ in and incorporates a control valve to maintain proportionality between the pedal and braking efforts. A feedback arrangement ensures a satisfactory feel to the pedal. Between the servo unit and the inlet manifold is a vacuum reservoir with a non-return valve, to give full braking power if the engine stops.

Normal B.M.C. unitary methods are followed in the construction of the body. The side members run the full length of the shell and are welded to the floor, which is extensively ribbed and embossed for stiffness. At the front is a sturdy, fabricated cross member that carries the suspension. As on the 15/60, the Farina-styled lines are characterized by a shallow roof pressing, a large boot, tail fins and a small peak over the rear light. The frontal grille, of traditional Wolseley shape, is deeper than on the $1\frac{1}{2}$-litre model, so that the bonnet line is above the crowns of the front wings, instead of dipping between them. Because of the greater breadth of the body in proportion to its height, its appearance is better balanced than that of the smaller car. The improvement is particularly noticeable from the rear, since the wider boot is better suited to the height of the fins. These fins have a slightly convex, substantially vertical rear edge, instead of the rearwardly raked straight line of the 15/60 model.

Both the windscreen and the rear light are of the double-curvature, wrap-round type with similar but opposite rakes. Like the side lights, they have stainless steel surrounds. The doors are wide, particularly those at the front, which are no less than $36\frac{1}{4}$ in between the hinge and shut pillars; the Wilmot-Breeden Zero-torque locks incorporate safety locking devices, which, when in operation, prevent the doors from being opened from the inside. Each side light has a hinged ventilating panel, and the main glass panels wind right down into the doors. The side lamps embody the flashing direction indicators and are positioned below the headlamps, which are supplemented by matched long-range and fog lamps let into the front skirt-panel. Both bumpers are taken round the quarters and carry overriders of greater depth than those of the 15/60.

The front seats are of the divided bench type, with individual adjustment but a common folding central armrest. There is ample leg room in the rear compartment, and the rear seat also has a folding central armrest. An additional armrest is provided on each door, except that adjacent to the driver. Leather is employed for the upholstery of the seats. The facia panel is of American burr walnut and the instruments are grouped on two circular dials, one on each side of the steering column. On the top edge of the facia is a resiliently padded roll, which is repeated on the edge of the parcels shelf beneath it. In front of the passenger is a lockable glove compartment, and another parcels shelf is incorporated behind the rear seat squab. Heating and demisting equipment is included in the standard specification, and there is provision for the installation of a radio. The spare wheel is housed in a separate compartment beneath the floor of the boot.

The kerb weight distribution of the Wolseley 6/99 is 54·6 per cent on the front wheels and 45·6 per cent on the rear wheels. For the 6/90, the figures were almost identical, 54 per cent and 46 per cent. The ratio of wheelbase : mean track is 2·02 : 1 and, because of the considerable length of the boot, that of wheelbase : overall length is 0·575 : 1. To get the generous seating width, the lateral overhang is relatively large, and this is revealed by the ratio of 0·78 : 1 between the mean track and the overall width of the car.

Cable Testing Set

TO ENABLE all types of wiring installation to be tested for earth leaks up to 3 megohms, Calidec Ltd., 12 to 15 Lupin Street, Birmingham, 7, is producing the Type MB portable cable testing set. The set can also be used for continuity tests and has the advantage over the normal equipment that the indication of an earth leak or an open circuit is given audibly and visibly, by means of a bell and indicator lamp system. There are in fact, two sets of such indicators so that testing for both kinds of fault can be investigated simultaneously.

The set is battery operated, to give a quick warming-up time, and its circuit embodies a thermionic valve and two high-resistance relays. A pressure of 90 volts is applied to the circuit under test, and the use of a high resistance, in conjunction with this voltage, ensures efficient operation. The dimensions of the apparatus are $10\frac{1}{4} \times 6\frac{7}{8} \times 7\frac{1}{2}$ in and, since it only weighs 9 lb with batteries, it is readily portable.

Electronic Tachometer

AN ELECTRO-OPTICAL tachometer, the Mayer Touro-Strob instrument, is to be marketed in Great Britain and Northern Ireland by Eric H. Bernfeld Ltd., 282, Kingsland Road, London, E.8. It can measure rotational speeds and cyclic movements over the range of 200 to 15,000 r.p.m. in three stages: 200 to 900 r.p.m., 900 to 3,600 r.p.m. and 3,600 to 15,000 r.p.m. As its name suggests, the Touro-Strob operates on the stroboscopic principle and can be used, if desired, as a true stroboscope, with a maximum frequency of flashing of 250 c/sec.

The source of the flashing light is a neon discharge tube, and its sharply defined red beam can easily be observed in daylight. This instrument is intended for operation on an A.C. supply of about 240 volts and 40 to 60 c/sec, and its consumption is 25 to 35 watts. The calibration is stated to be accurate, after a stabilizing period of 5 minutes, within $\pm\frac{1}{2}$ per cent. Details are available from the concessionaires.

THE HALLSWORTH MOULDING SYSTEM

Basic Automatic Units, for Either a Complete Foundry or the Mechanization of Existing Facilities, Developed by Rubery, Owen & Co. Ltd.

In the production of light ferrous and non-ferrous castings in medium or large quantities, traditional foundry conditions have changed but little, except as regards the health aspect, in several decades. Production efficiency has been raised to some extent by the introduction of machines for moulding, squeezing, and stripping, and by the use of mechanical conveyors. In general, however, such improvements would appear to be simple extensions of conventional methods. Major causes of inefficiencies in the moulding department, it has been suggested, are limitations of floor space and the fact that moulders are not infrequently required to carry out subsidiary duties, such as pouring, knocking-out, and sand conditioning.

The Hallsworth system represents a new approach, based on the use of relatively small, easily handled, moulding boxes and fully automatic equipment where desirable and economically justifiable. Flexibility is retained in the system so that a number of different pattern plates can be used within a single shift without requiring more adjustments than can be undertaken by a semi-skilled mechanic. Equipment is arranged about an automatic moulding machine that sprays, charges with sand, edge-tucks, squeezes, removes surplus sand, strips, and deposits the mould on a conveyor track ready for assembly. This sequence is carried out on a four-station machine served by a number of ancillary appliances keyed to the operational sequence.

Historical background

The moulding machine was originally conceived in 1951 as a result of method study in a jobbing foundry. It was early realized that to employ the machine effectively, box positioning, sand charging, and half-mould removal operations would need to be synchronized. A working model was designed and constructed and with this sufficient experience was gained to justify the production of a working prototype. This machine performed satisfactorily in production trials but quickly it became obvious that it would be necessary to evolve a comprehensive scheme incorporating the ancillary equipment for the machine to be used to maximum advantage.

A system was designed theoretically, and the firm of Rubery, Owen and Co. Ltd., Darlaston, South Staffordshire, then undertook the development of the machine and the complete project. In view of the departure from conventional practice it was deemed advisable to obtain extended operational experience before the system was introduced to general industry. Accordingly, several installations were placed in production in iron foundries for proving trials, and from the practical experience gained minor improvements were made to the machine and the controls. A MK.II machine, embodying modifications incorporated after trials in service, is shown in Fig. 1.

The basic principle was proved to be sound and, in its final, developed form, the solution of the problems was on the following lines:

1. The foundry process to consist of three interlocked cycles of operation:
   (a) Boxes—from empty, through moulding of cope and drag, closing over, clamping, pouring, cooling, shake-out, and return to moulding machine
   (b) Metal—from melting, casting, shake-out, dressing, and return of scrap to melting
   (c) Sand—from shake-out, through reconditioning, moulding, pouring, and return to shake-out.
2. The production of two basic machines, designed for cyclic operation, one for moulding and one for casting. To link them with standard or special equipment to provide the sand cycle.
3. Because of the friable nature of the green mould, to provide supervision at certain stages. Such labour to be used at operations where a high degree of fatigue has been eliminated and the operator cannot materially influence the rate of material flow.
4. The system to be adaptable, so that a complete "packaged foundry" could be installed in a bare shop or, alternatively, selected items could be provided for a foundry where ancillary equipment of a suitable size and speed order was available.

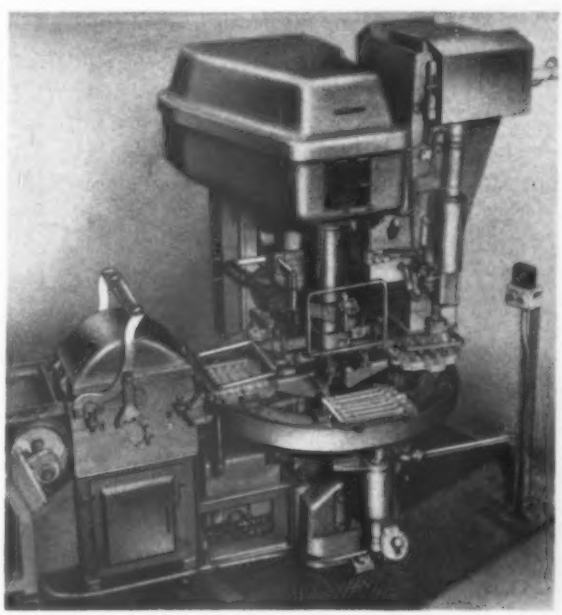
Moulding boxes

A relatively small moulding box is essential in order that the complete cycle of operations can be arranged on equipment of a reasonable size. Fast-moving equipment is cheaper to produce and simpler to operate than large, ponderous plant. Experience shows that in the manufacture of small castings, a small rapidly moving box yields a greater output per man than a large box. Size in a moulding box adversely affects the time cycle by reason of operator fatigue and larger, heavier mechanical handling equipment.

The size of box selected for commercial production, Fig. 2, is 12 in × 9 in × 5 in. Determining influences were:

1. Half-boxes to be precision made and interchangeable.

Fig. 1. The Hallsworth four-station automatic moulding machine, MK.II



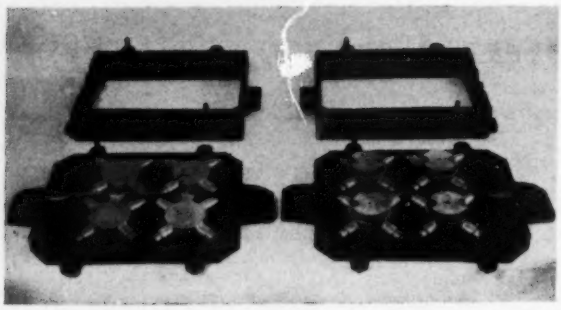


Fig. 2. Two half-boxes, 12 in × 9 in × 2½ in, and their associated pattern equipment

2. Boxes to be designed for vertical pouring, with its associated benefit of smaller running system and higher casting yield.

3. A small box implies easy handling. Where manual operation is economically desirable, operator fatigue is reduced.

4. Pattern-making costs are less for small boxes than for large boxes. The small pattern plate is more easily changed.

5. It is the rate of turnover of boxes which controls the rate of output, rather than the size of the box.

6. A small box tends to prevent the occurrence of "sand slip".

7. With various shapes of castings it is possible for the small box to maintain the ratio of area of castings to area of box at a high level.

General layout

The schematic diagram in Fig. 3 is a layout of a complete plant designed according to the foregoing principles. The four conveyors—sand supply, coring, box return and underground sand return—have been arranged to provide the necessary linkage between the three main cycles of sand, metal, and box movement. These, together with the provision of compressed air and electricity supplies, enable the system to function as a complete foundry unit.

As a result of the policy of producing a compact, efficient moulding unit, the complete plant can be accommodated on a concrete apron approximately 30 ft × 60 ft. No arrangements have to be made for heavy plant, and an overall thickness of 6 in of concrete is quite ample. Because of the flexibility of plant, the particular size suggested is not mandatory and the length:width overall plan ratio could be increased or reduced according to specific requirements. The size suggested, however, gives a reasonably economical plant using the minimum lengths of conveyors.

Transfer of spent sand and its elevation to the reconditioning plant necessitates an underground conveyor pit approximately 30 ft × 5 ft × 6 ft deep. This is connected to a pit approximately 10 ft square × 7 ft 6 in deep, in which are housed the bucket elevators. The magnetic separator necessitates a separate pit 5 ft × 3 ft × 4 ft deep. Under normal conditions the foundation pits are lined with a 6 in thick layer of concrete.

Automatic moulding machine

Based on machine-tool construction principles, the machine comprises a rigid frame to which are attached the mechanisms to carry out the different functions. The moulding table rotates between the base and head castings which, structurally, present a self-contained force system. Control and actuation are simple, and relatively complex electronic systems have been avoided. The machine has four stations and a rotary table which automatically conveys the half-box and pattern from station to station. At station 1 the operator locates the box part on the pattern plate and pulls down the hinged sand-allowance frame on to the box part. The table indexes to station 2 to receive sand and in its arc of movement is subjected to a spray of parting fluid. A semi-rotary hopper turns through 180 deg, discharging sand into the box part on the table. A frame fitted to the sand chute then moves downwards, to "edge-tuck" by compressing the sand round the edge of the mould.

Fig. 4. Moulding machine and sand-feed conveyor. To the right can be seen the guard of the magnetic separator

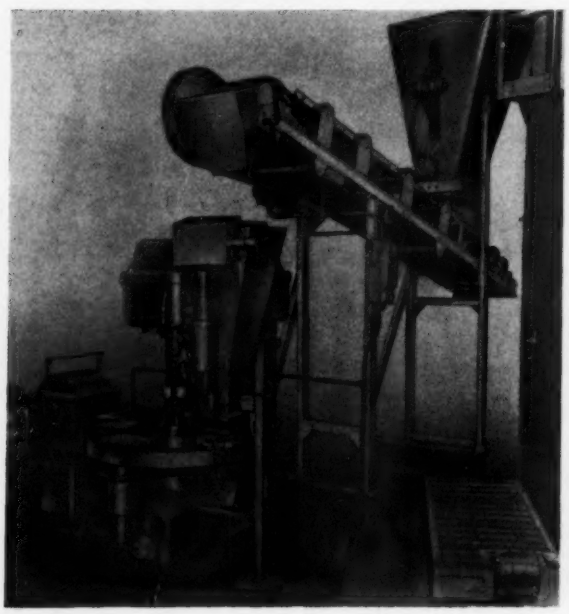
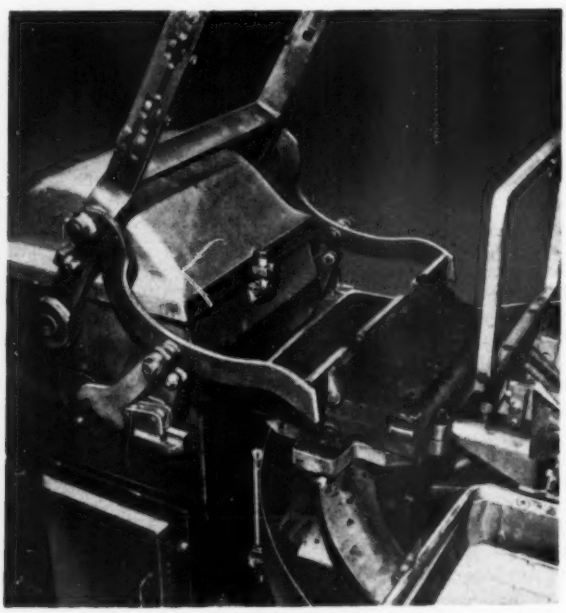


Fig. 5. Surplus sand is cut from the mould by a scraper frame. On completion of its stroke the mould lift operates



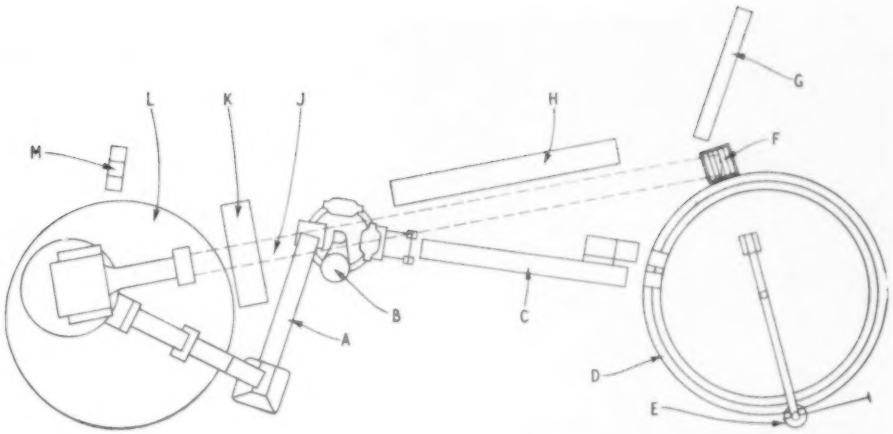


Fig. 3. Schematic diagram of layout of a typical plant. Floor area approximately 60 ft × 30 ft

A *sand-feed conveyor;* **B** *automatic moulding machine;* **C** *coring conveyor;* **D** *rotary mould conveyor;* **E** *pouring mechanism;* **F** *shake-out;* **G** *casting delivery conveyor;* **H** *mould box return conveyor;* **J** *sand-return belt conveyor, underground;* **K** *magnetic separator;* **L** *sand-reconditioning plant;* **M** *control panel*

The table next indexes to station 3 for the main squeeze operation. During this movement the sand-filled box part passes under an adjustable sand-levelling device (the "static strickle bar") which strikes off surplus sand. While moving from station 3 to station 4, the hinged sand-allowance frame is restored to the vertical position to enable the scraper bar to strickle the mould. The pattern plate is vibrated and the mould stripped, to be received by the take-off arms, which by now have moved into position. The cycle is completed when the take-off mechanism places the half-mould face upwards on a moving belt conveyor. It will be appreciated that all the operations described take place simultaneously, and four boxes and patterns are in use at any instant.

It is necessary for an operator to be present, but his sole functions are:

1. Visually examine pattern plate and clean with compressed-air gun if necessary.
2. Position box part.
3. Bring down sand-allowance frame.

None of these operations are fatiguing to any great extent and all the essential moulding operations are automatic.

Operation and Control

The table-indexing mechanism is used as the master control, determining the operating cycle and thus ensuring that no overlapping of operations can occur. Precise control of operation speed is obtained, where necessary, by mechanically linking a hydraulic dashpot system to a pneumatic power system. Motive power for table rotation is supplied by a pneumatic cylinder, its speed controlled by a hydraulic cylinder, for example. Overall cycle time is controlled electrically by process timers; individual cylinder speed is controlled pneumatically by speed control valves.

Motion is transferred from the index cylinder to the rotary table by a pawl and indexing cam. The pawl is mounted in a carrier which oscillates around the indexing cam, thus imparting the necessary interrupted rotary motion. Movement is under the control of a hydraulic cylinder ensuring that starting and stopping the table is reasonably shockless. During the power stroke of the index cylinder the pawl is in the engaged position, so that motion is carried directly from the cylinder to the machine table by way of a pair of indexing plates and a vertical shaft. The shaft is mounted in upper and lower bearings and has the pattern table keyed to it. Adjustable cams are attached to the indexing plates which control, in sequence, the primary ancillary operations which are performed when the table is in motion. The operations which are undertaken when the table is still, are under the direct control of a pilot valve actuated from a cam on the mould take-off arms.

An emergency, pedal-operated, solenoid stop control is fitted externally to the base casting. Depression of the pedal cuts off the supply of air to the machine and unloads all cylinders, thus bringing all motion to a standstill. As a consequence of this the indexing mechanism is unloaded and the table can be moved freely by hand.

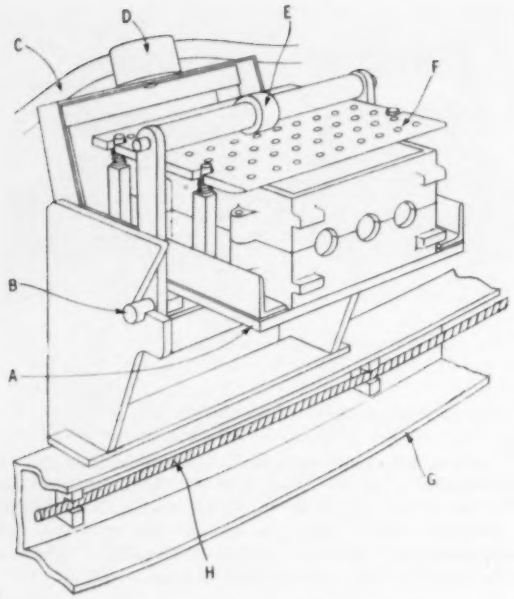
Machine table

Running between the cartridge-type bearing assemblies fitted to the base and head castings, is a 3½ in diameter hollow steel shaft, keyed to which is the fabricated steel, four-station, moulding table. The table is designed to limit any surplus-sand accumulation by using an open structure, the principal members of which are shaped to a form designed to prevent sand adhesion.

Also fitted to the table driving shaft is an assembly of four hinged frames, so sprung that they occupy either vertical or horizontal positions. They are vertical during loading and unloading and horizontal when edge-tucking and squeezing. At the end of the cycle of mould production the frame is vertical, and one of the operator's functions after placing a new moulding box in position is to lower the frame, where it is held down horizontally by over-centred springs. Between the squeezing station 3 and mould-ejection station 4, a projection at the back of the frame passes under a positioned wheel which forces the frame into the vertical position. The purpose of the frame is to contain surplus sand for ramming requirements and to prevent spillage during these operations.

Sand metering and edge rucking

A steel chute is fitted to the moulding machine, seen generally in Fig. 4. At the top of this chute is mounted a double-sided hopper comprising two opposed, box-like compartments, one of which is always in position to receive sand from the belt conveyor working in conjunction with the reservoir hopper. The double-sided, semi-rotary hopper is mounted horizontally on bearings attached to a squared shaft. At each indexed operation, the hopper turns through 180 deg, in either clockwise or anti-clockwise direction, emptying the sand-filled compartment. The amount of sand entering the hopper between two indexing operations is controlled by adjusting the gate on the reservoir hopper, thus controlling the level of sand on the belt. The steel chute is fitted with a movable frame at its discharge outlet. This frame has four "keep" sides, the leading edges of which are flanged to squeeze the sand at the edge of the moulding box and thus perform the function of "edge-tucking". Vertical movement of the "edge-tucking" frame is effected by two double-acting pneumatic cylinders. The overall dimensions of the frame provide a working clearance



A *base plate*; **B** *pivot*; **C** *cam rail*; **D** *tilt roller*; **E** *clamping-pressure roller*; **F** *spring-supported top plate*; **G** *track outer ring*; **H** *wire-rope drive*

Fig. 9. The mould-clamping pallet with a moulding box in position

between the mould and the frame, enabling the squeezing force to go into the body of the sand.

Main squeeze

The main squeeze cylinder is accommodated within the head casting and operates vertically. Its piston rod carries a serrated press plate of suitable dimensions to allow entry into the sand frame when squeezing. In order to accommodate the heavy thrust which occurs, two pads positioned on the underside of the table are arranged to contact on a pair of tapered supporting rollers on the base casting, so stopping table deflection and limiting main bearing wear. Any variation of pressure required, can be obtained by adjustment of an easily accessible control on the air line.

Mould stripping

The lifting mechanism consists of an aluminium casting on which are mounted four ejector pins positioned to locate at the corners of the box. Vertical movement of the unit is obtained by a specially designed heavy-duty air cylinder, to which two steadying hydraulic cylinders are coupled mechanically. The mounting structure is a bracket cantilevered from the base casting. Before the four pins are in contact with the box, a standard pneumatic vibrator comes into action on the underside of the pattern plate and continues during lifting, preventing sand sticking to the pattern and thus facilitating a clean stripping action.

Take-off and scraper

The mould take-off, shown in Fig. 6, consists of two steel lifting arms, semi-rotatably mounted in self-aligning roller bearings located on a steel frame. The arms are actuated by two trunnion-mounted cylinders, positioned in parallel, an air cylinder supplying motive power and a hydraulic cylinder controlling speed. The linear motion of the cylinders is transposed to angular movement by a rocker arm connected to the cylinder nose ends. Lifting arms are fitted with spring-loaded catches which engage corresponding projections on the moulding box. This method of attachment prevents any movement of the half-mould relative to the arms during its transfer from the machine to the coring conveyor, and yet allows it to be withdrawn smoothly by the conveyor.

The scraper mechanism is designed to reproduce mechanically what is essentially a hand operation under normal circumstances. During this operation excess sand is removed from the mould back by the cutting action of the scraper bar or blade. It is essential that after the scraping operation the level of sand should not protrude above the top edge of the moulding box. A frame is fitted over, and pivoted to, the mould take-off lifting arms. In the moulding box discharge position, the scraper rests on stops with the leading edge facing towards station 4. An adjustable, steel cutting blade is attached to the leading edge of the frame. Rollers are set flush either side of this scraper

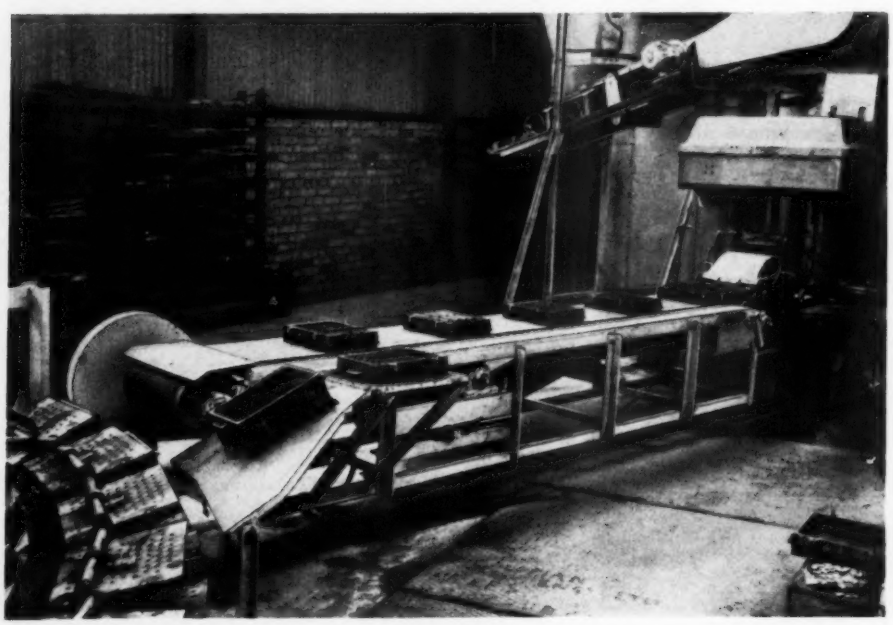


Fig. 7. Coring conveyor with half-moulds in transit. A closed mould is positioned on the slipway ready to load into a pallet on the rotary mould conveyor for final clamping

blade, designed to run along the opposing faces of the moulding box sides. When the take-off arms move to engage a mould, they are preceded by the scraper frame and the semi-rotary movement of the arms is converted into a linear movement of the scraper frame. Thus the scraper blade passes across the moulding box cutting away excess sand, as shown in Fig. 5.

Coring conveyor

A general view of the coring conveyor, Fig. 7, shows the half-moulds after discharge from the take-off mechanism progressing towards the rotary track. Simple core laying is carried out while the moulds are in motion and the half-moulds then proceed towards the clamping mechanism. If more complicated coring is required, a semi-circular table is positioned to the side of the coring conveyor. The operator pulls the matching half-moulds on to the table, lays the cores, closes over the cope mould, and restores the completed mould to the conveyor.

Rotary mould conveyor

To ensure efficient labour utilization, and for inspection purposes, the completed mould is loaded manually into the rotary conveyor, Fig. 8. The effort needed is slight, since a slip is provided at the end of the coring conveyor down which the boxes slide into the pallet. From this point the mould is handled mechanically.

The mould-clamping pallets, Fig. 9, are carried on the outer ring of the rotary track. Each unit consists of a base and a spring-supported top plate, and is mounted in pivots about the bottom pallet between two upright supports. In each pair of pivot members two links are fitted loosely, one each side of the mechanism, and connected by a steel bar which projects a short distance through each link. This bar is provided with a pressure roller in the centre. A roller is also fitted to the underside of the mechanism and engages with a cam rail controlling the angle of tilt. When the mechanism is tilted, clamping is effected by the projecting ends of the link tie bar interfering with the top of the two vertical supports, causing the pressure roller to force the top plate downwards on to the mould.

When the mechanism is restored to the horizontal position again by the control of the roller engaging with the cam rail, the clamping pressure is relieved. The springs exert an upward thrust to the top plate, and so facilitate the removal of the mould. The pallet mechanism is so designed that any unevenness of the sand face does not interfere with the ultimate contact of both box parts and the mechanism is aligned essentially about the major pouring axis of the box.

Fig. 6. Take-off station of the moulding machine. The half-mould is engaged by the take-off arms and is being transferred to the coring conveyor. The box may be withdrawn if the coring is complicated

Following the casting of the mould, the track continues to take the pallet round with no change in angle. During this period the cast metal is solidifying and steps have been taken to ensure that no vibration occurs. The speed of the track is controlled to enable the casting to solidify completely, and this in turn controls the whole clamping, pouring and shake-out cycle. The track is wire-rope driven from an electric motor through a variable speed gearbox.

After the appropriate cooling period has elapsed, the

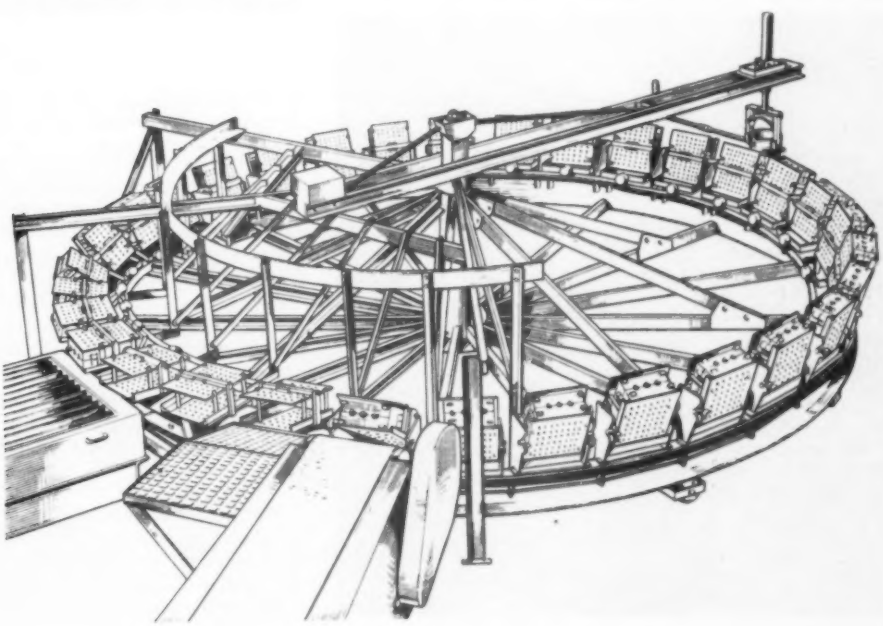


Fig. 8. Rotary mould conveyor and swinging-arm pouring mechanism. Shows the tilting of the mould pallets for vertical pouring

poured moulds are automatically restored to the horizontal position and unclamped ready for removal and the shake-out operation.

Pouring

To pour the moulds in the vertical position, it is advisable that the ladle lip be kept as close as possible to the "down-gate", and the ladle height be adjusted about its lip during the pouring of the metal. It is also essential that the heavy fatigue associated with this operation be eliminated, and consequently the simple mechanism that can be seen in Fig. 8 was evolved.

Standard foundry ladles are used but the carrying handle is modified. The ladle and carrying handle are supported in a self-adjusting cradle suspended from a swinging arm. The cradle consists of two hooks and is supported by a compression spring working in a housing which, in addition to catering for the rotary and vertical movements, also allows the ladle to be moved radially inwards or outwards. The arm from which the mechanism is suspended rotates about the centre of the mould conveyor and has a balance weight fitted at the opposite end to facilitate rotation and to eliminate operator fatigue. The ladle handle is provided with two trunnions, one on each side of the ladle, which pivot within the two hooks of the cradle. The cradle support spring is constrained to equal the combined weight of the assembly and consequently it adjusts its height as the metal poured reduces the total weight.

The shake-out is a conventional vibratory, frame-type machine located over the underground sand-return conveyor. The box parts are placed on the gravity roller conveyor, Fig. 11, and returned to the moulding machine.

Sand processing

Used sand is carried by the underground conveyor beneath the magnetic separator, shown in Fig. 4, to remove any foreign ferrous metal. It is then discharged into a bucket elevator and raised to a 3 ft diameter rotary screen above the storage hopper. A schematic view of a typical batch installation is shown in Fig. 12. Located beneath the storage hopper is a measuring hopper for discharging a given quantity of sand into the mixing mill.

From the mill the sand is discharged into a disintegrator, where it is broken down to a free-flowing consistency. It is then raised once more to the reservoir hopper which supplies the sand-feed conveyor of the moulding machine. The hopper is provided with an adjustable gate which is used to control the rate of sand input into the semi-rotary hopper of the moulding machine. The reservoir hopper and its associated conveyor form one of the independent units of the system. It is important to ensure that the semi-rotary hopper of the moulding machine is filled uniformly, and consequently the discharge end of the conveyor is fitted with a reciprocating sand chute driven from the head pulley shaft, see Fig. 4. This ensures that, irrespective of the depth of sand on the belt, it is distributed evenly in the hopper. Whichever type of sand-processing installation is provided, the sand cycle is controlled automatically by electrical sequence gear.

Control panel

The many electrical devices in the complete installation must be in sequence with the instigator of movement—the moulding machine. The central electrical control panel houses "direct-on" motor starters for bucket elevators,

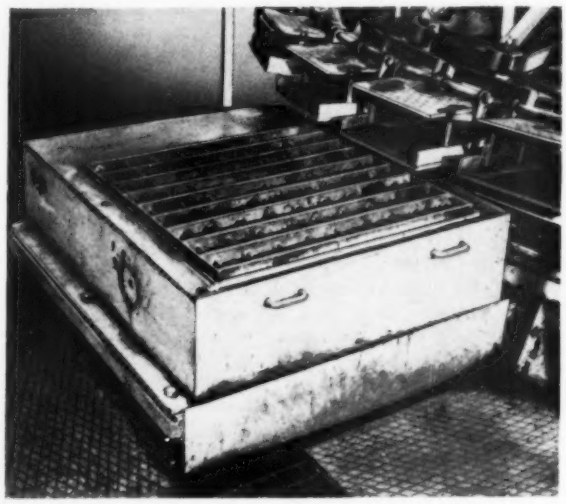


Fig. 10. Above: A pallet in the discharge position adjacent to the vibratory mould shake-out

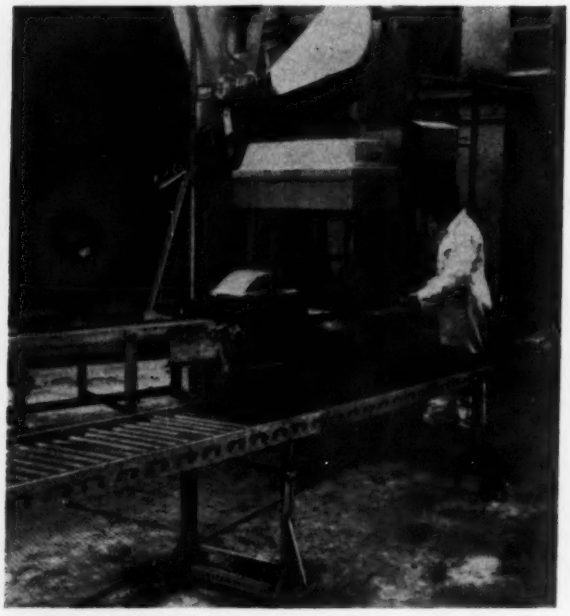


Fig. 11. Left: An operator placing a box part on the moulding machine. In the foreground is the box-return conveyor

elevator feed conveyor, rotary screen, magnetic separator, sand return conveyor, shake-out, mould conveyor and core conveyor. There is also a star-delta starter for the sand mill, a contactor type switch controlling the rectifier which supplies D.C. for the magnetic separator, and a control circuit transformer.

At the top of the panel are mounted two indicator lamps showing that the control circuit and rectifier are energized. The circuit is so arranged that all the motors must be started in their correct sequence. This prevents a "pile-up" occurring in any part of the plant, but for test and maintenance purposes a switch is fitted which allows any motor to be started in any order. This switch is behind the panel.

Flexibility

The moulding machine is capable of producing at least 240 completed moulds per hour. Depending on the output from the moulding machine, the type of metal or alloy, and the section of the casting, the speed of the rotary mould conveyor track is variable to give a cooling period of from

4 to 8 minutes. There are instances, however, where it is possible to reduce the cooling period considerably as, for example, when very small, light castings are being produced.

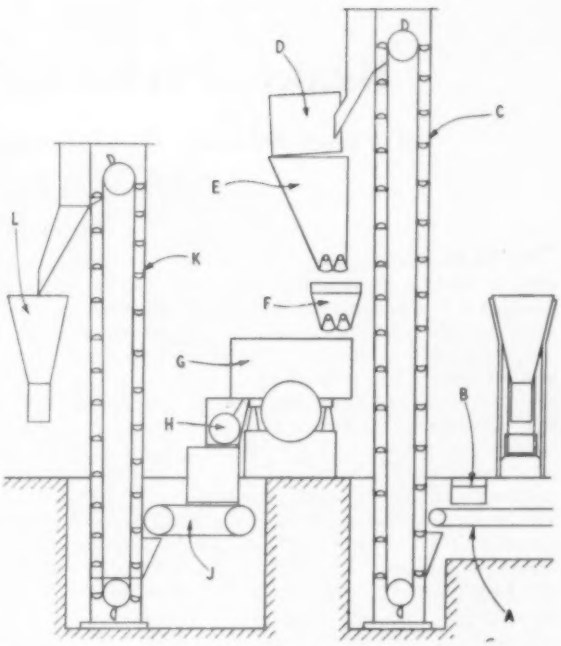
It is not always necessary for a foundry to have all the plant described. Many foundries are equipped with adequate sand reconditioning facilities and it is only necessary to ensure that a steady, controllable flow of sand in the correct condition reaches the rotary hopper on the moulding machine. Again, though the plant described has been arranged to give reasonable line flow of the basic moving elements of sand, metal and moulding boxes, it can, nevertheless, be grouped to suit almost any shape of shop and to meet any particular local requirements. All of the conveyor tracks, for example, can be made available in any length or shape, without altering the basic concept.

Economics

When considering the replacement of manual or semi-mechanized methods by a unit of this type, a foundryman would naturally use the results obtained in his own foundry as a basis of comparison. In general terms, the smaller foundries find it difficult to obtain an accurate cost assessment as manual moulding includes several ancillary operations, the performance of which may be carried out in some instances by several operators and in other cases by the moulder himself. Such operations include the supply and preparation of sand, positioning of cores, box collection, metal pouring, and knock-out of the casting. For this reason, a direct comparison of the two methods is difficult as any cost figures chosen for manual operation would not cover all cases likely to be encountered.

Some idea of the degree of comparison may be obtained by considering that with a five-man team operating the unit, a rate of at least 240 completed moulds per hour may be obtained with an output per man per day of the order of, say, 1,000 lb. This is a tentative figure assuming that fairly light castings of the order of 2½ lb per box are being produced. In a number of cases, on working installations, the casting yield per box has been as high as 8 lb, this excluding, of course, the weight of the running system. A representative group of castings produced on a Hallsworth installation is shown in Fig. 13.

In terms of manual operation, using a box holding 5 lb of castings (this normally implies a larger box than is used on the automatic machine—say 18 in × 12 in × 5 in to represent more conventional practice) one operator would have to produce 200 boxes per day in order to approach the conservative output, previously mentioned, of the automatic machine. In addition, he would have to carry out the attendant operations himself, including sand preparation, metal pouring and all material movement. A rate of production of this order is almost impossible and, even taking into account the variation from foundry to foundry, it might be deduced that the production per operator increases three to four times when using a unit of this type. The economic benefit of even trebling the output per man under present conditions of high labour charges, and the difficulty of obtaining skilled men, needs no emphasis and gives some idea of the rapidity with which the initial capital outlay might be recovered.



A sand-return conveyor; B magnetic separator; C bucket elevator; D rotary screen; E main storage hopper; F batch-measuring hopper; G mixing mill; H disintegrator; J elevator feed conveyor; K bucket elevator; L reservoir hopper

Fig. 12. Diagrammatic arrangement of sand-reconditioning plant

The illustrations shown as Figs. 12 and 13 are reproduced by courtesy of Platt Malleable Castings Ltd., and those in Figs. 7 and 11 are by the courtesy of Charles Perks Ltd.

Fig. 13. A range of malleable iron castings produced on a Hallsworth automatic moulding installation

Brake Performance Measurement

A Simple Detonator Device Developed by the Road Research Laboratory

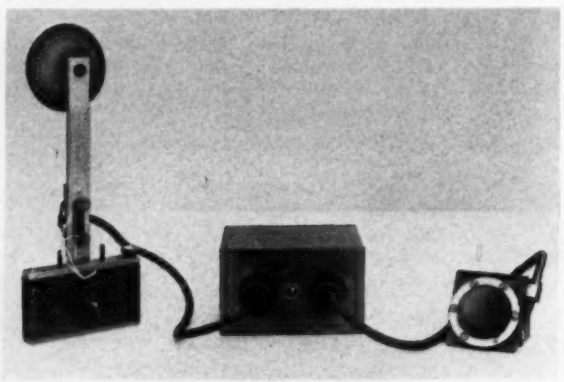
R. D. LISTER, B.Sc., A.M.I.Mech.E., A.M.I.C.E.

SEVERAL methods are in common use for the measurement of braking performance. Of these, the most readily accepted is to determine the braking distance from a known initial speed, but considerable care is needed if reliable results are to be obtained. The requirements are: accurate measurement of the speed of the vehicle at, or just before, the instant the brakes are applied, and a precise indication of the position of the vehicle at that instant.

It is well known that the speed of the vehicle can be measured with a fifth wheel[1] or by using precision laboratory equipment to determine accurately the time taken by the vehicle to traverse a short measured distance—usually between 2 and 5 ft—just before the point at which the brakes are applied. Alternatively, the vehicle speedometer can be calibrated and the reading noted when the brakes are applied.

Inaccuracy often arises, however, in the determination of the braking distance. Methods depending largely on personal judgment, such as those involving measurement from a fixed object at which the driver attempts to apply the brakes, have been shown to be inaccurate.[2] An alternative method, which obviates this inaccuracy, has been in use at the Road Research Laboratory for some years. It is based on the employment of a simple and inexpensive device capable of marking, on the road, the position of the vehicle at the instant the brakes are applied. With this device, a small electrically-fired explosive detonator shoots a chalk pellet onto the road surface as soon as the driver depresses the brake pedal.

The complete apparatus, as used at the Laboratory, is shown in Fig. 1, and details of the magazine and pedal switch are given in Figs. 2 and 3. Ten detonators and chalk



pellets are accommodated in the magazine, which can generally be held in position on the car by a suction cup. If a fairly flat, smooth surface on which to use the suction cup is not available, it may be necessary to bolt or clamp the magazine in position. The magazine should be fixed so that its lower edge is 6 to 9 in from the ground. Measurements have shown that, at this distance, the delay between closing the firing circuit and the chalk pellet reaching the ground is about 0·008 sec, which is equivalent to a distance

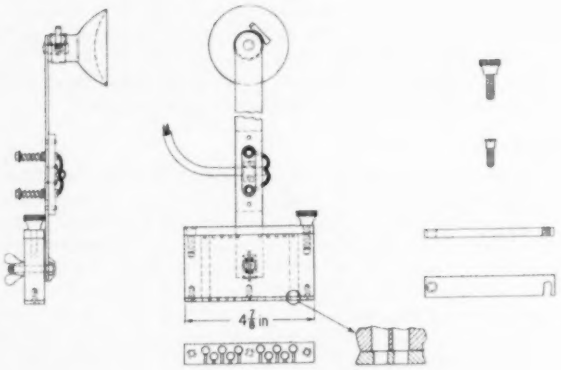


Fig. 2. Above: Detonator magazine and mounting

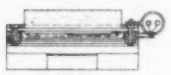
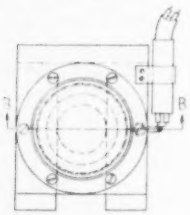


Fig. 3. Right: The brake pedal switch layout



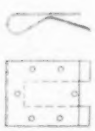


Fig. 1. Left: Detonator device complete with the battery box and the pedal switch unit

of only 4 in at 30 m.p.h., and can therefore be neglected. It is useful to include a hand-operated master switch in series with the battery, pedal switch and detonator, so that the circuit can be rendered inoperative until the last moment: this allows the brakes to be used normally, without firing the detonator, during manoeuvring before the test run.

It will be noticed that the contacts of the pedal switch are insulated from the body of that switch. This arrangement allows the vehicle battery to be used for firing the detonators and prevents the closure of the contacts from having any effect on other recording gear that may be operating at the same time. The detonator makes a fairly loud report on firing. Only a small mark is made on the road surface, and it is removed by the next fall of rain. The magazine should be cleaned occasionally, because a slight coating of wax is deposited in it when each detonator fires.

This device was developed as part of the programme of the Road Research Board of the Department of Scientific and Industrial Research and the article is published by permission of the Director of Road Research.

References

1. R. D. LISTER and R. G. STEVENSON: "Fifth Wheel for Measuring Speed," *Bull. Mot. Ind. Res. Ass.*, 1952, 4.
2. H. J. H. STARKS and R. D. LISTER: "The Measurement of Braking Distances," *Bull. Mot. Ind. Res. Ass.*, 1950, 4.

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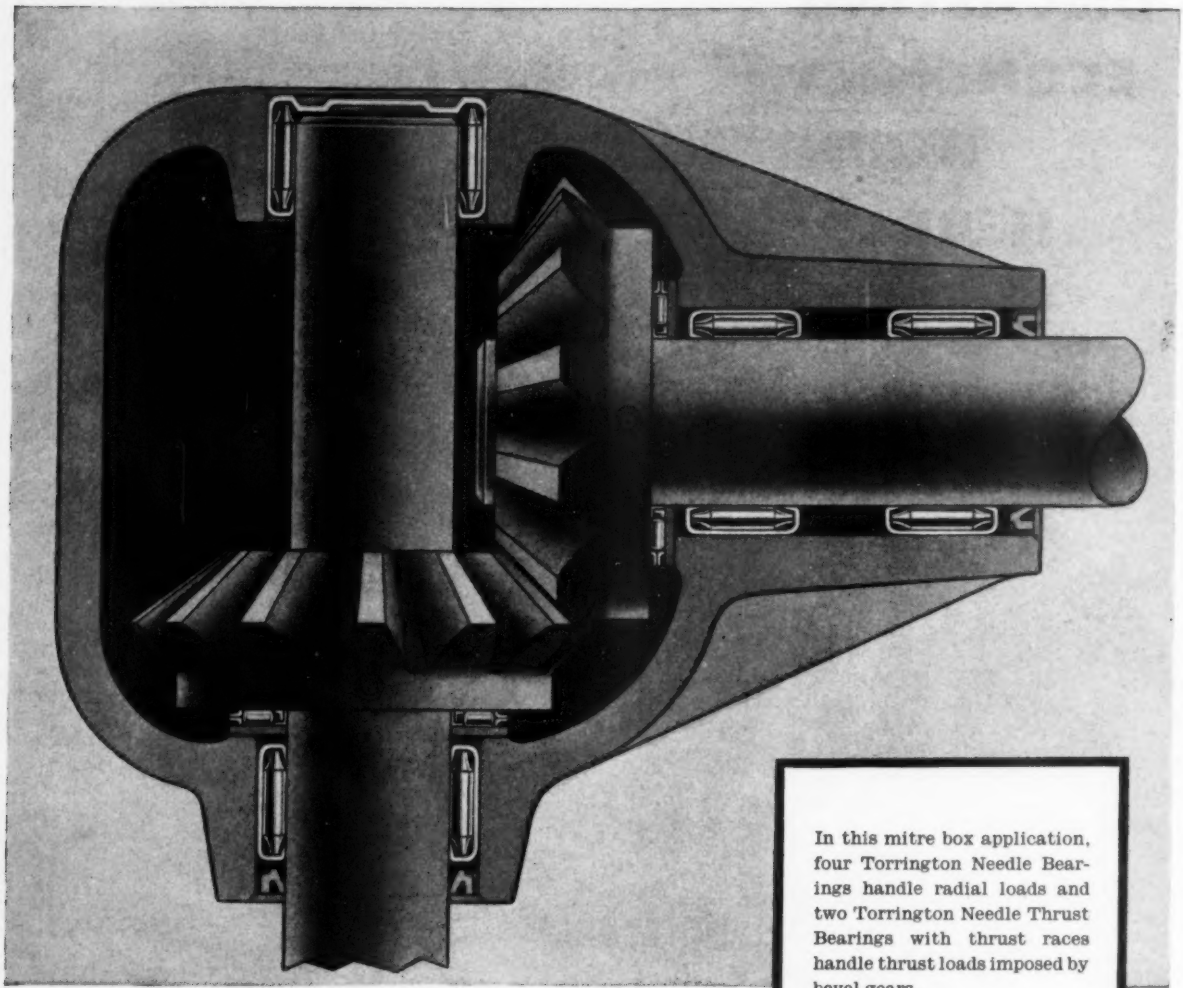

S.59.D

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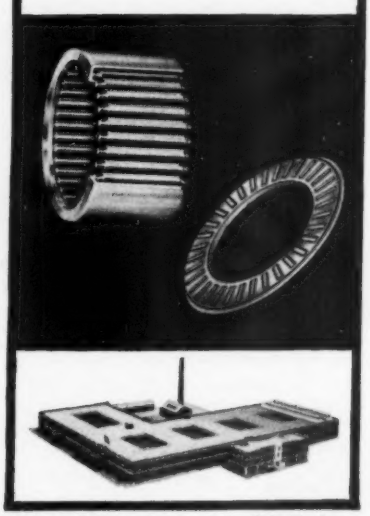

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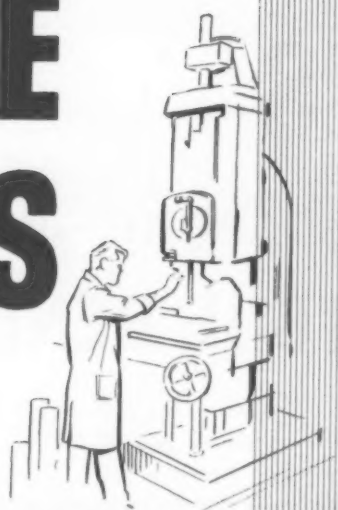

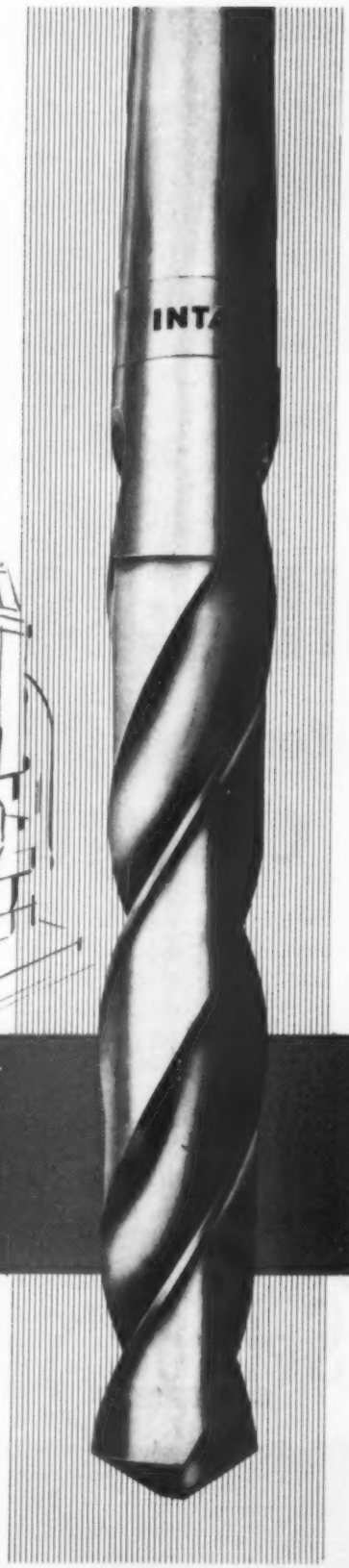

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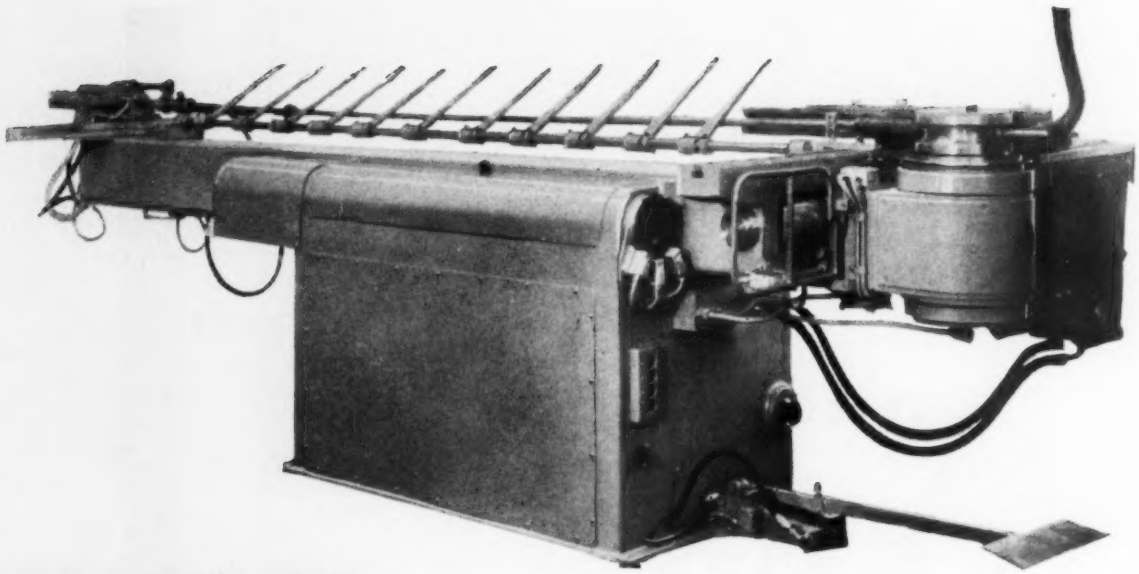

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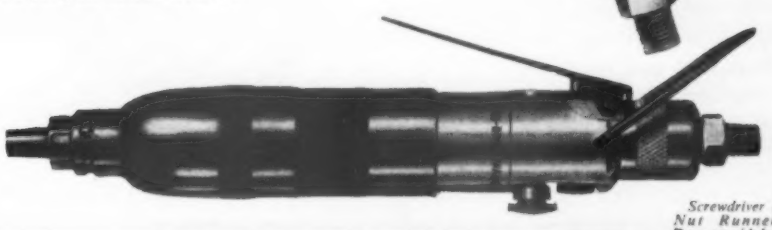

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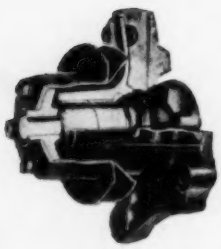

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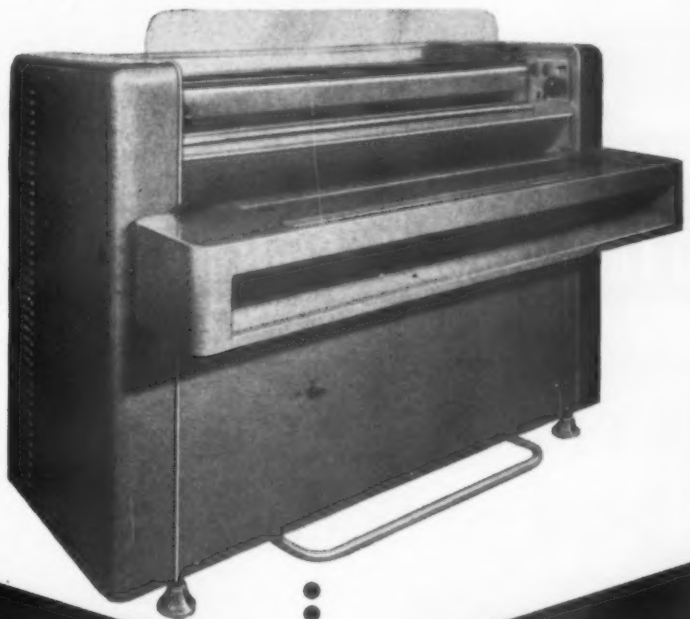

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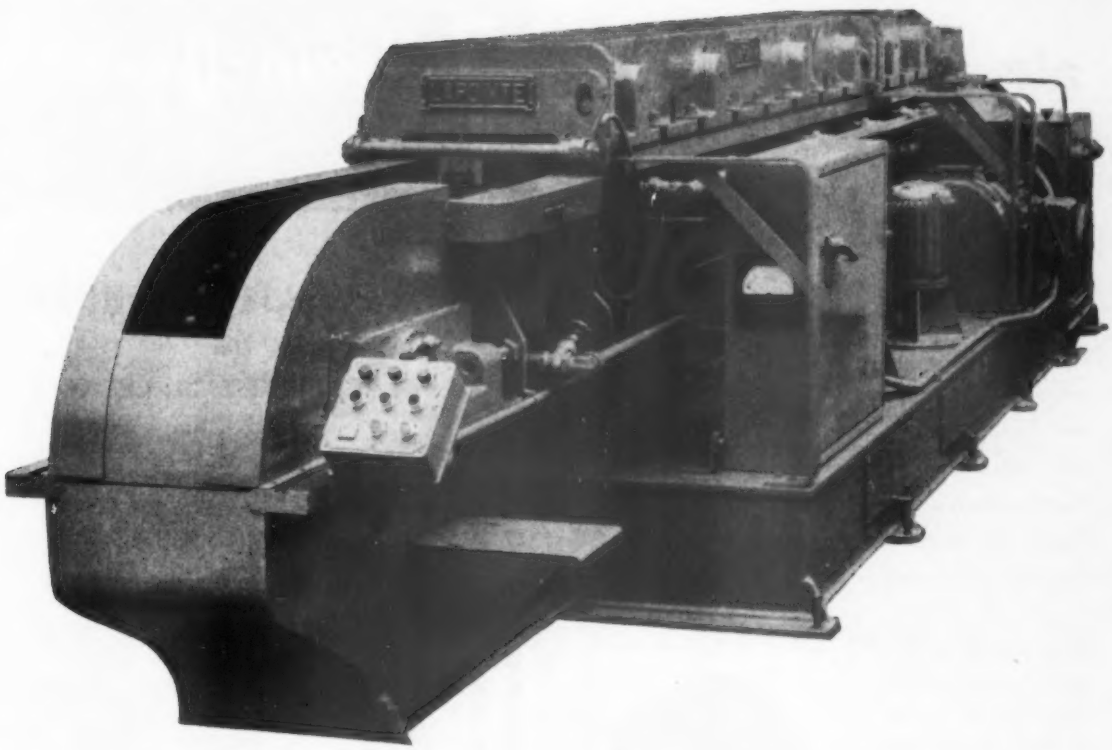

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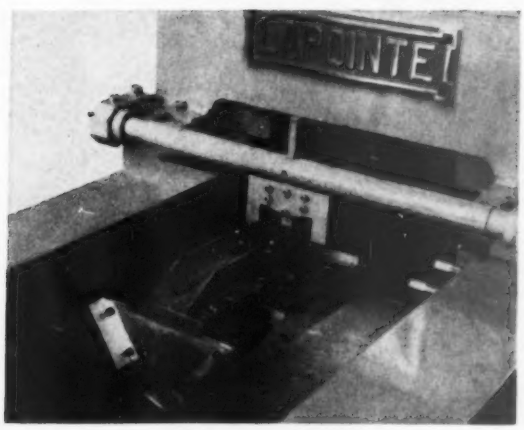

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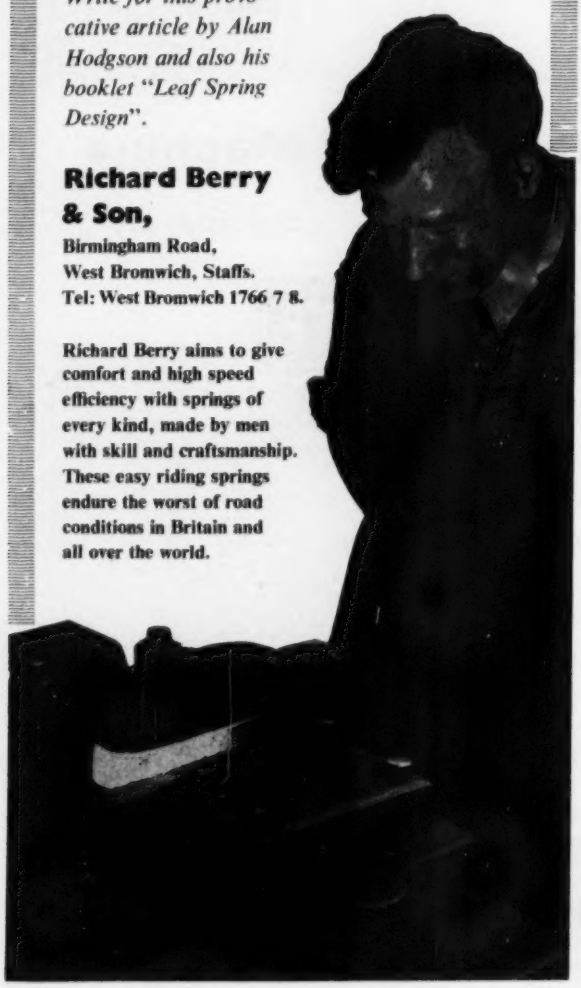

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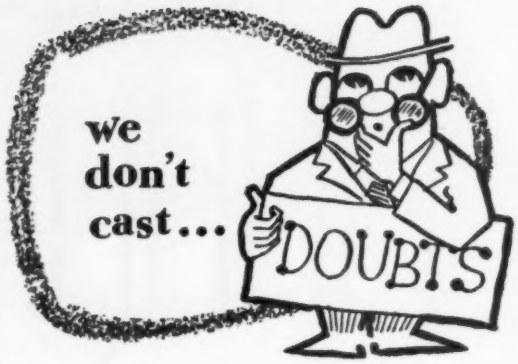

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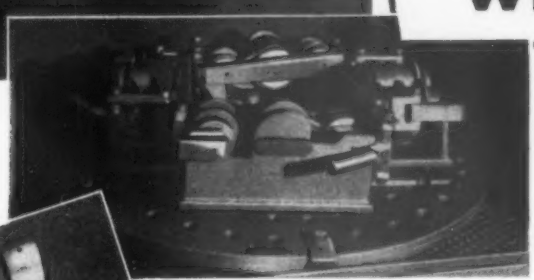

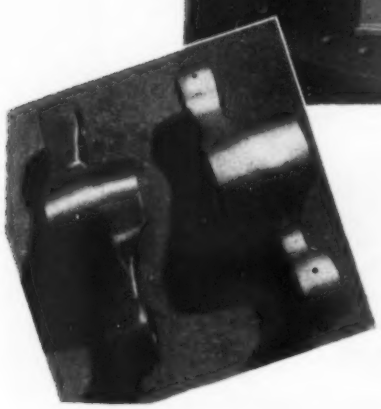

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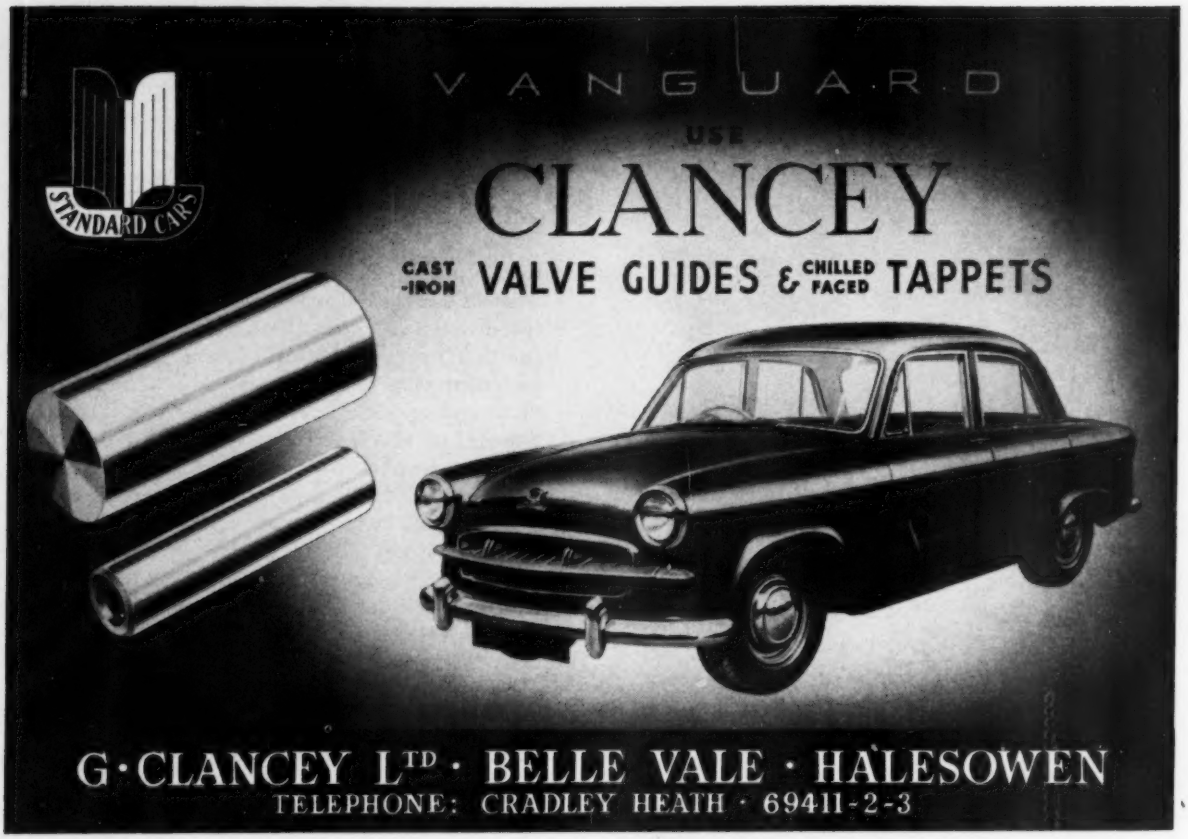

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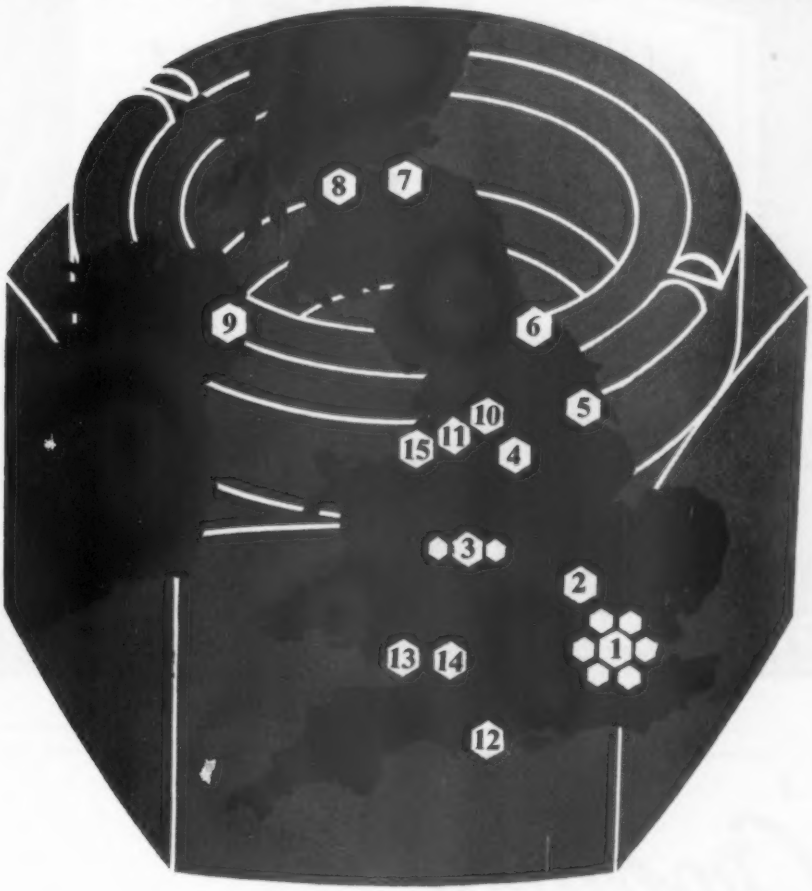

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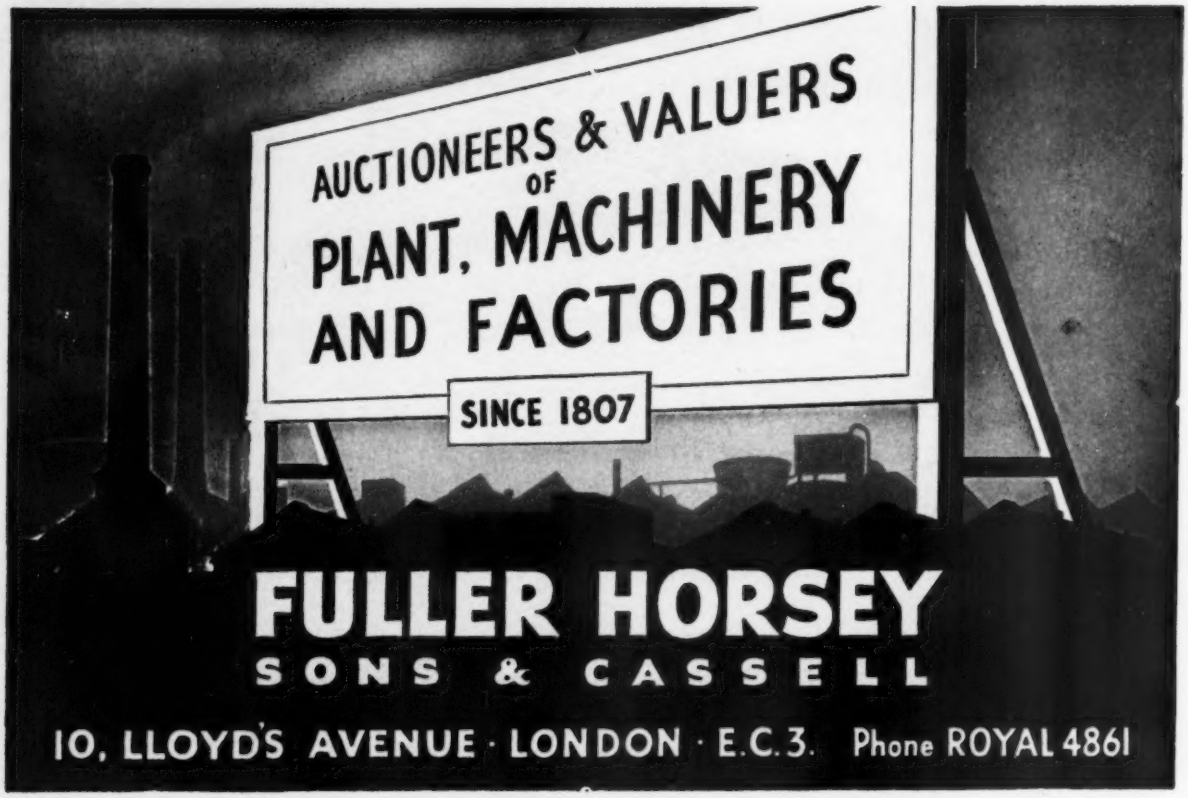

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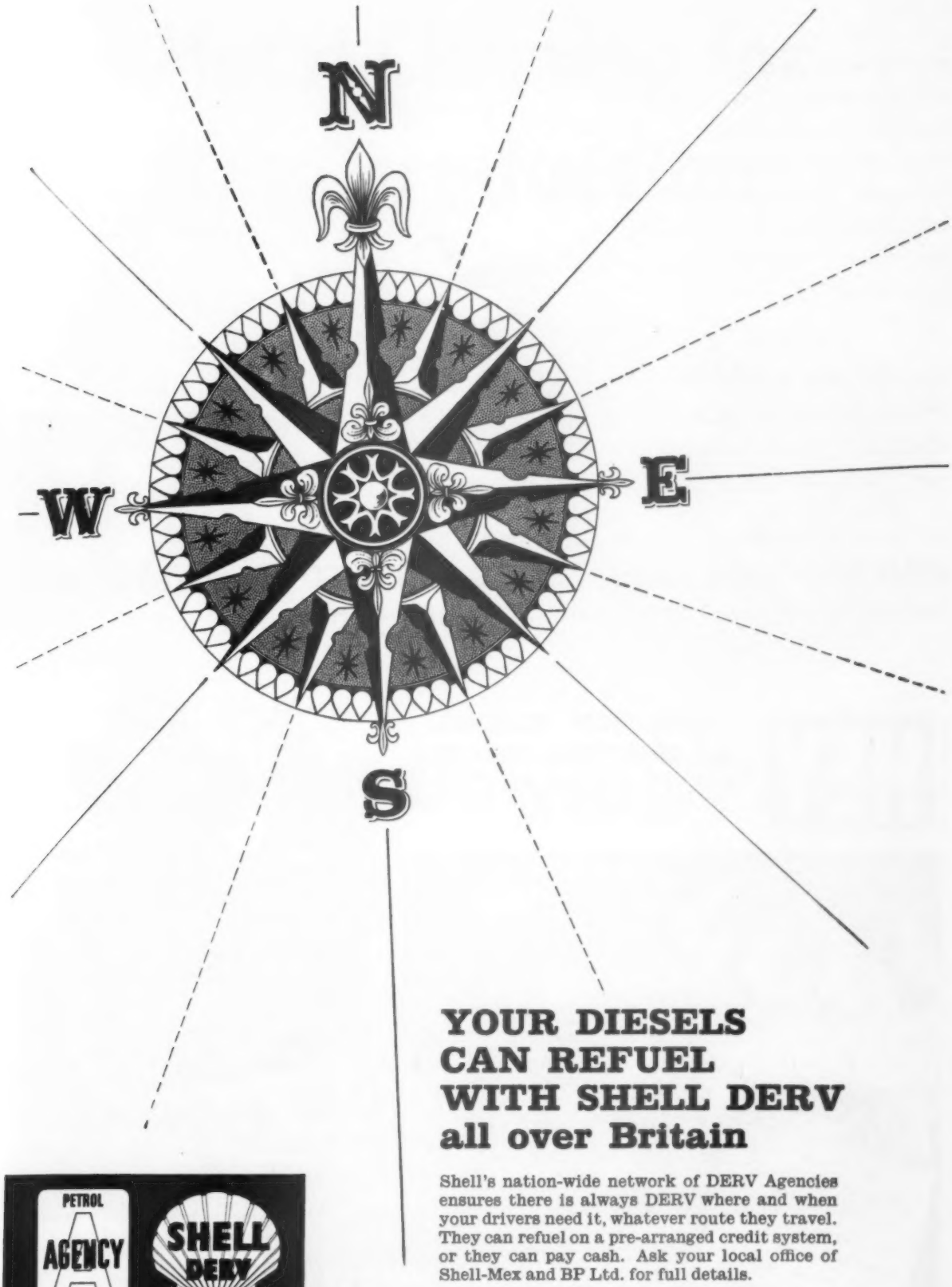

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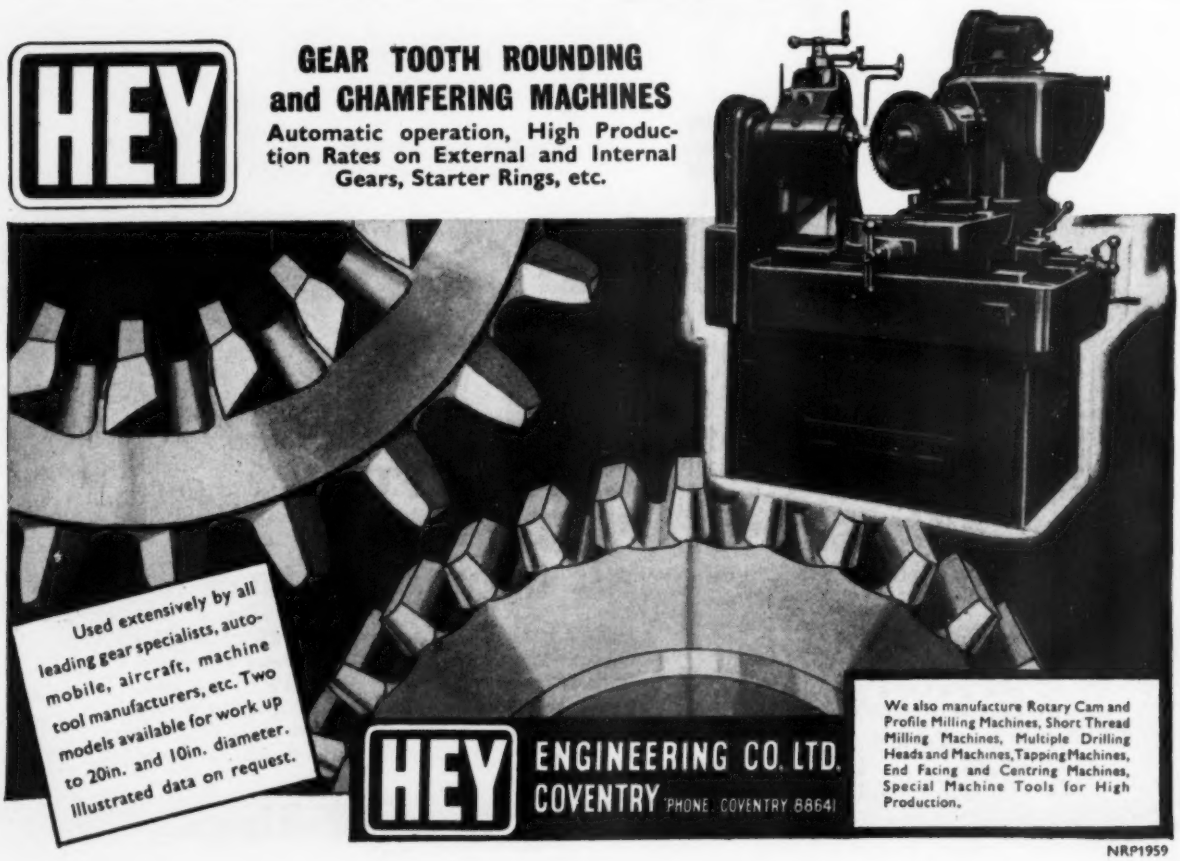

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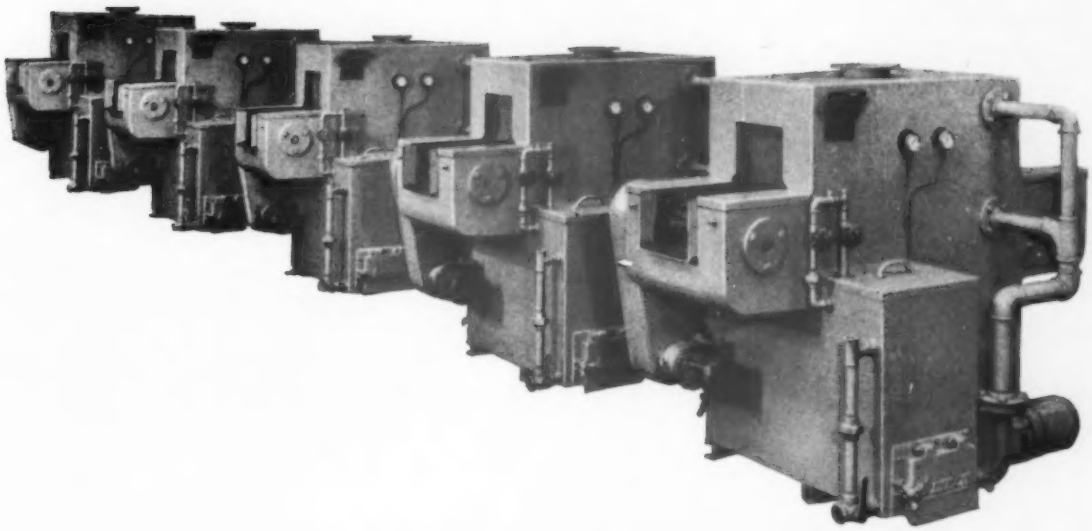

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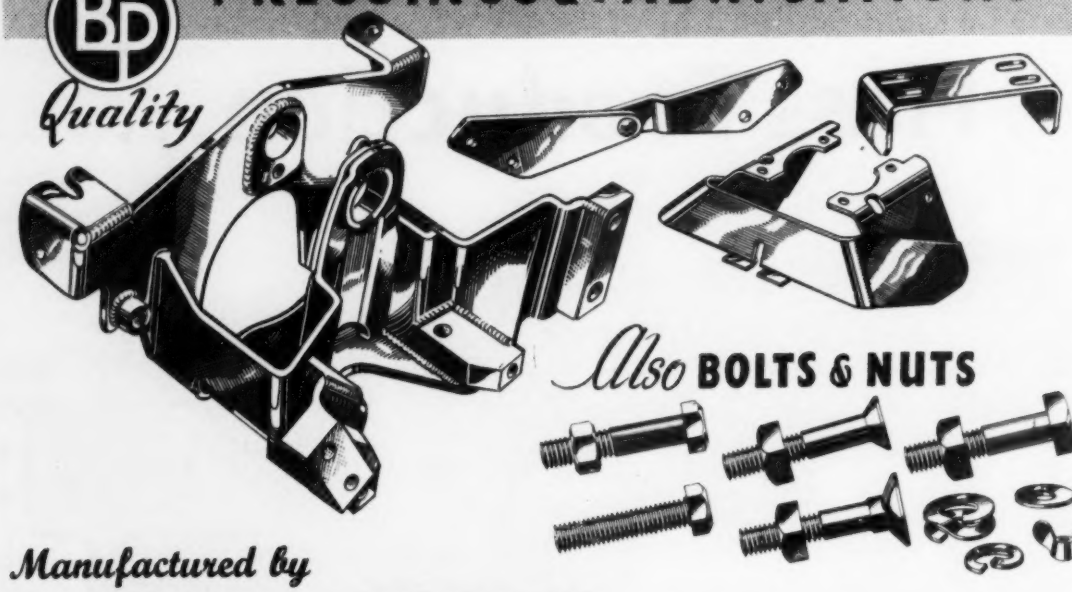

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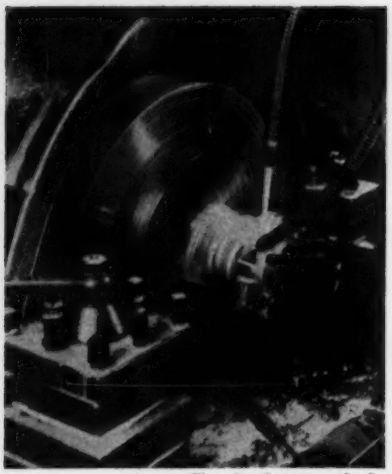

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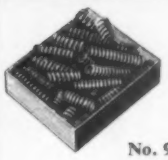

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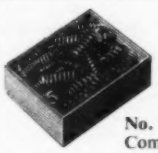

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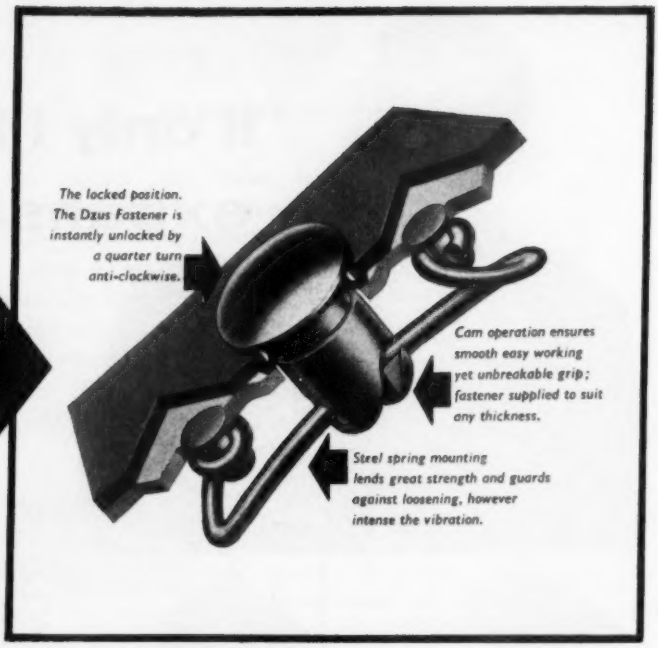
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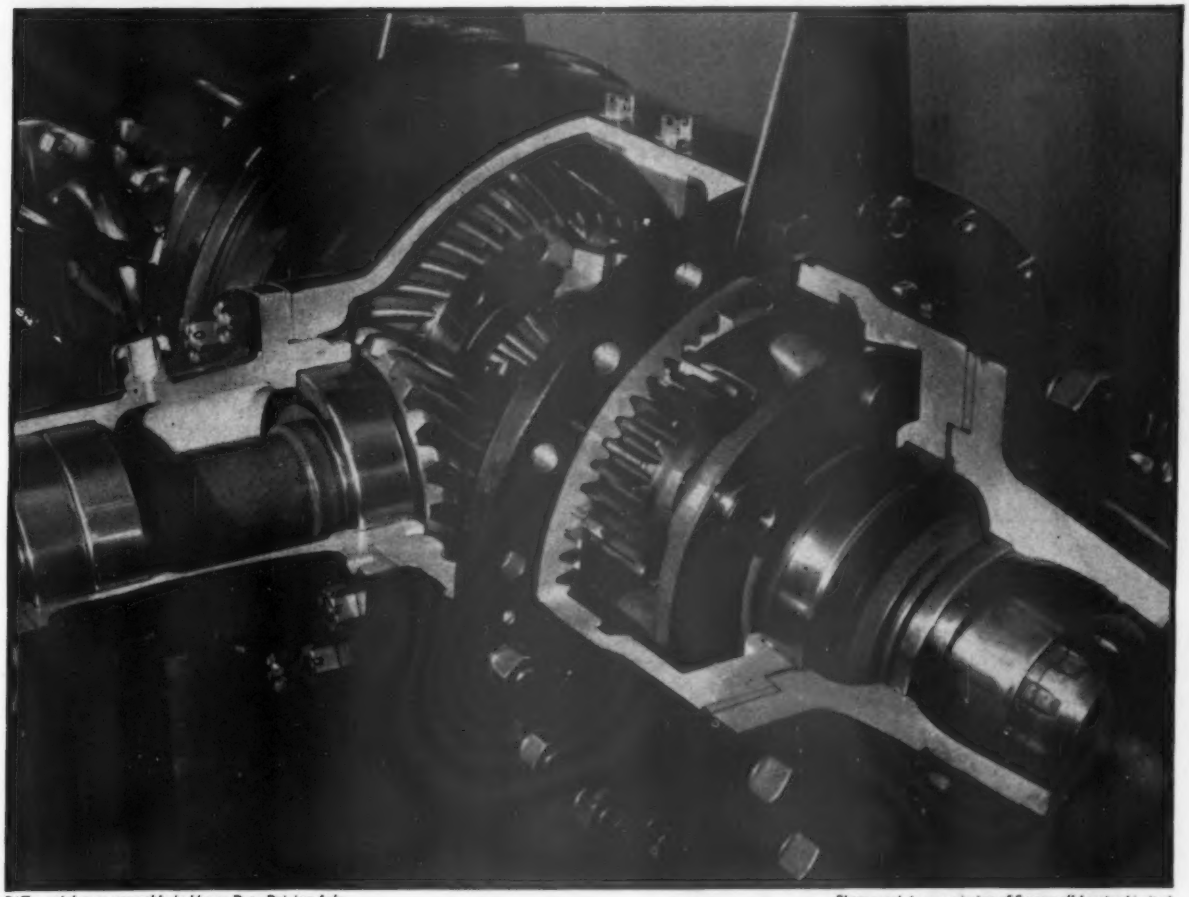

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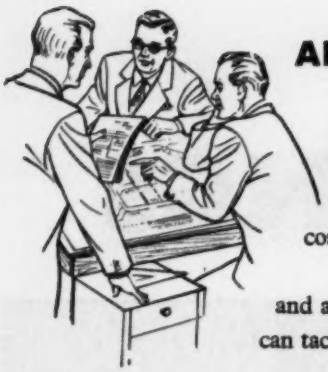

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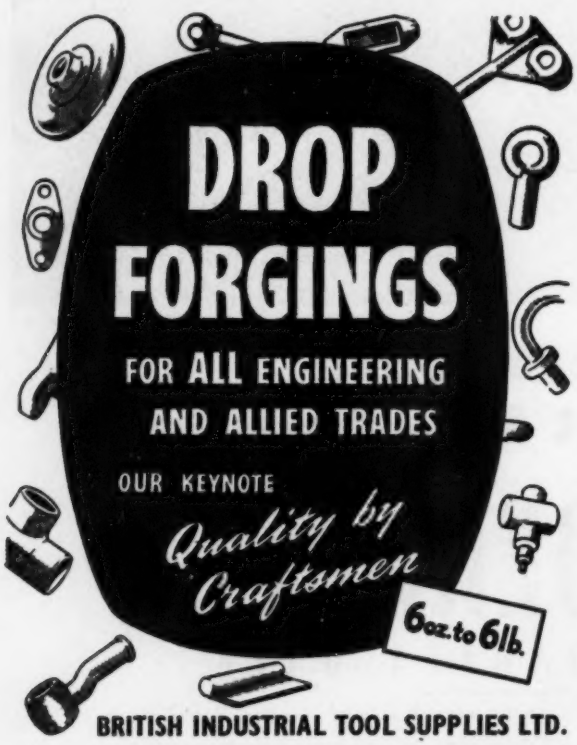

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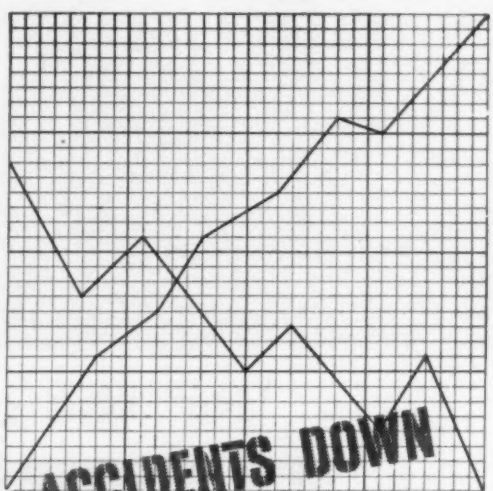

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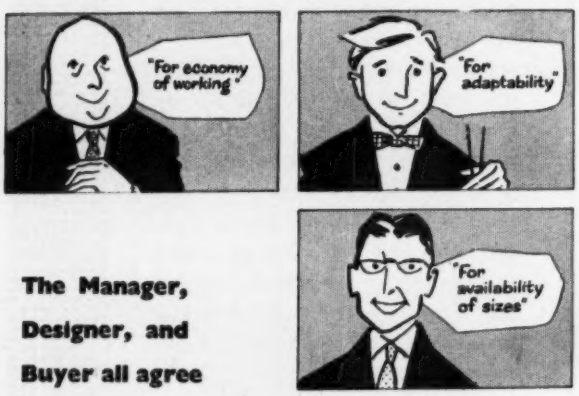

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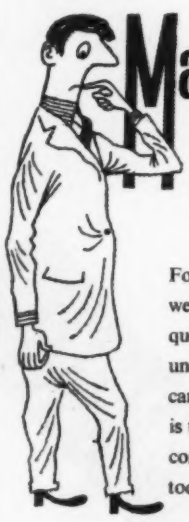

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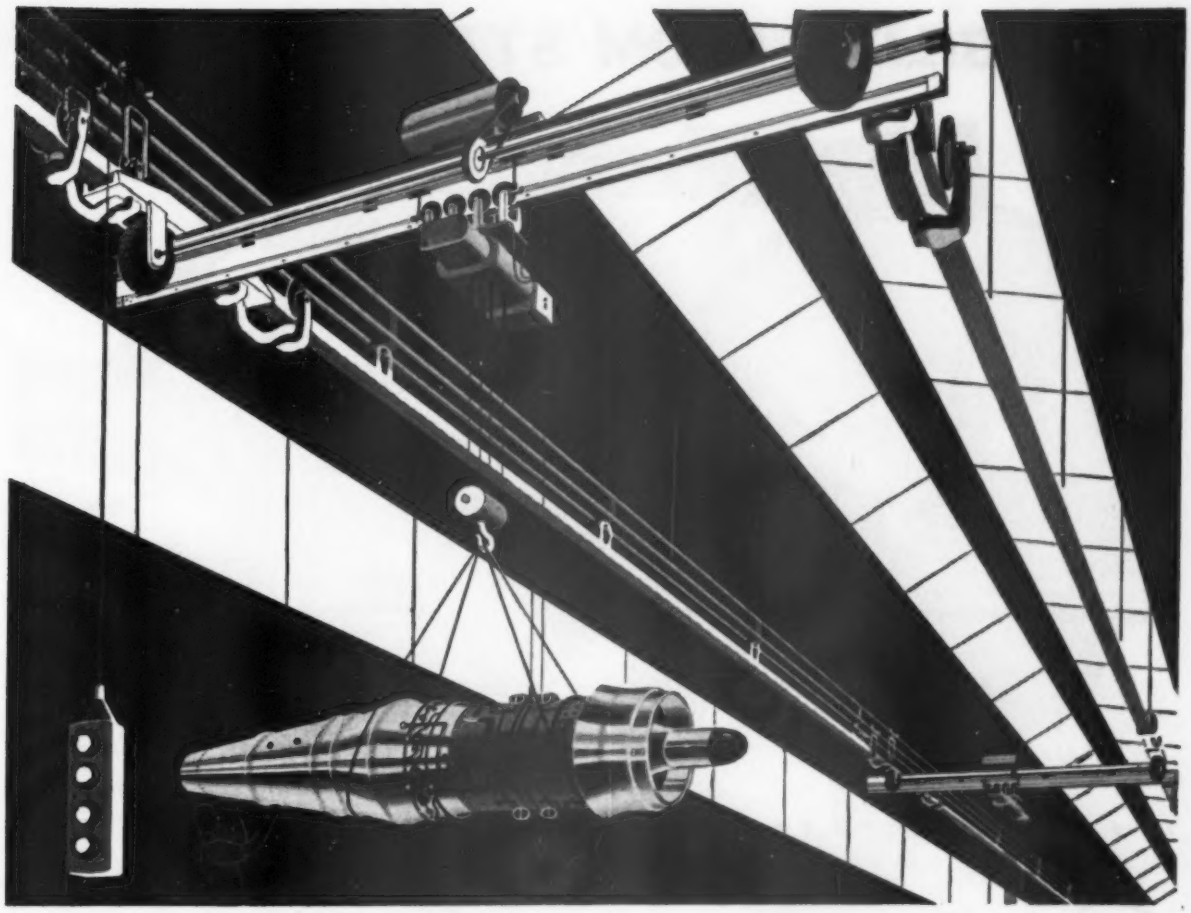

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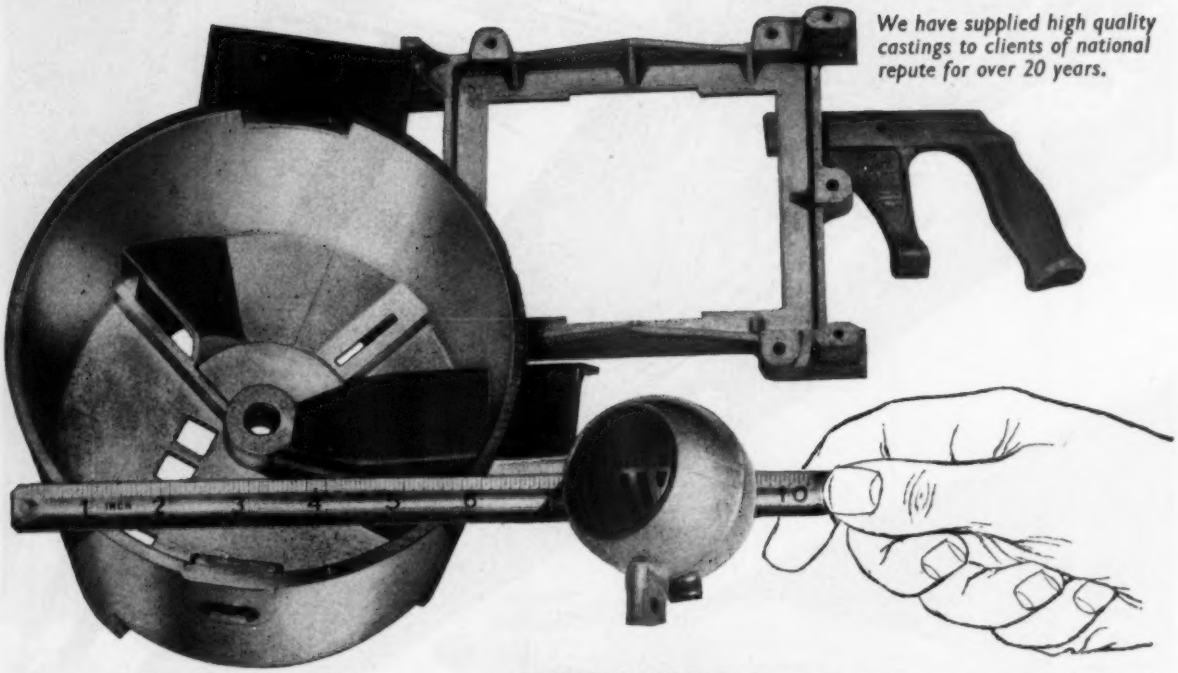

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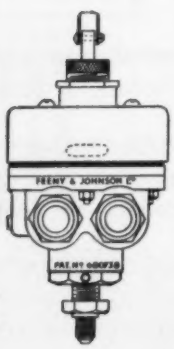

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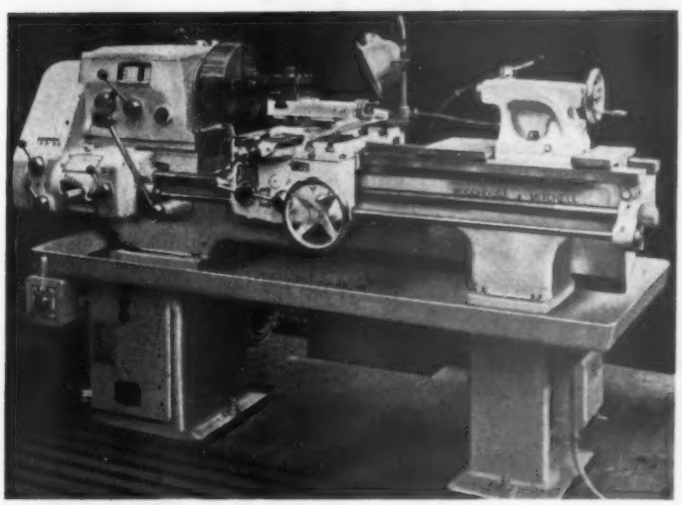

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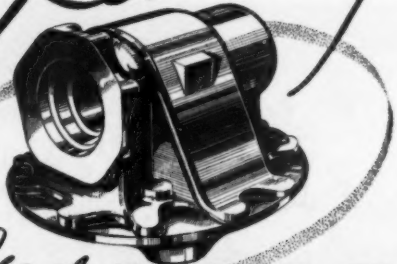

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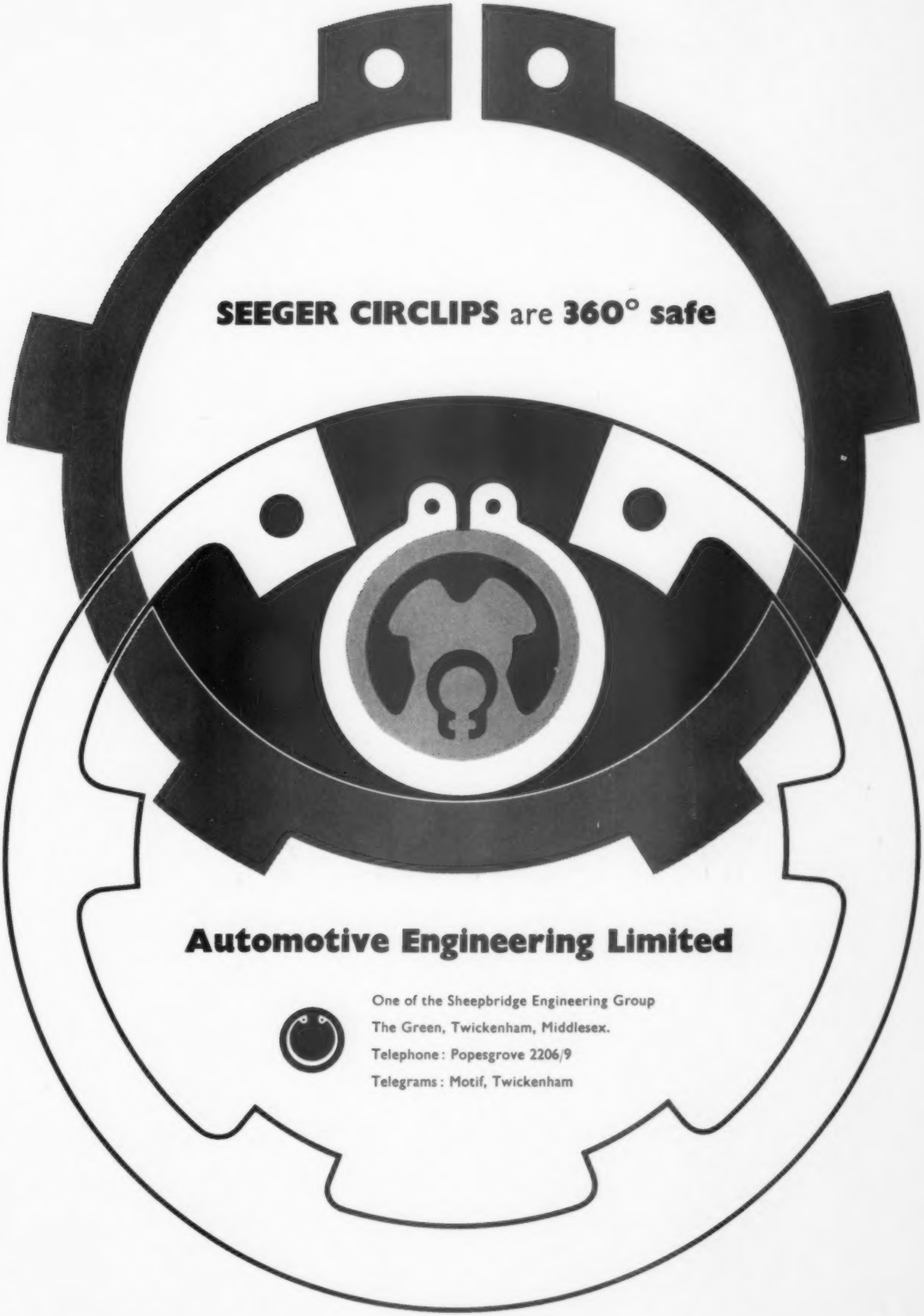
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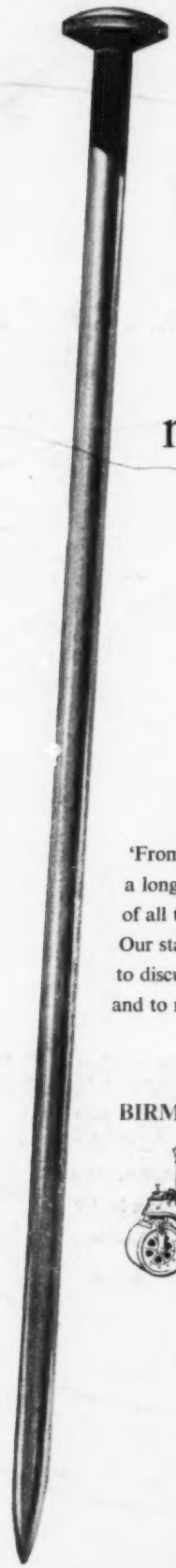

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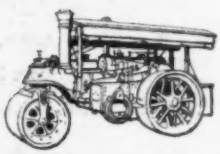

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